AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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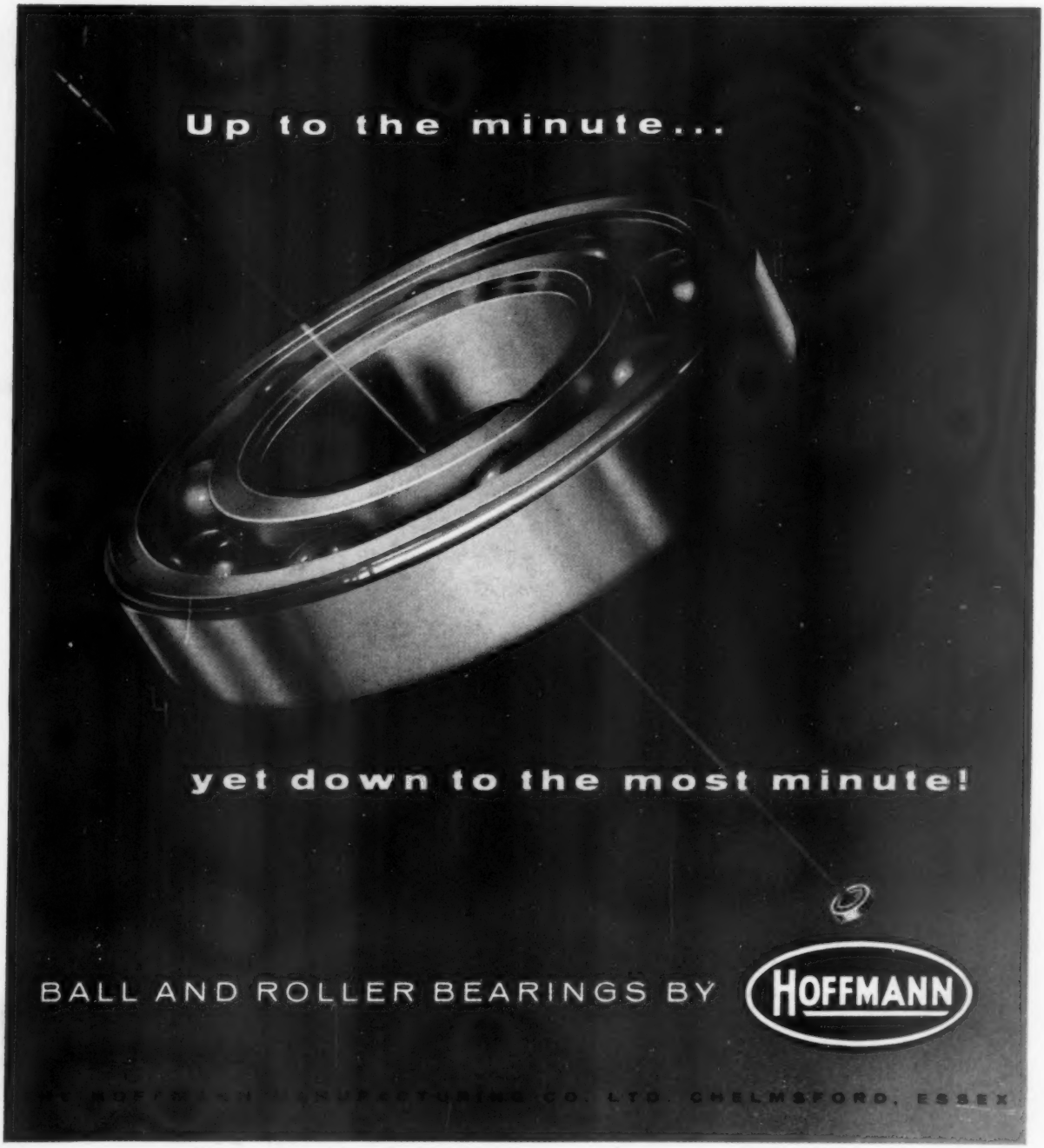

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–in NPK Hexagon Socket Screws

GKN engineers employ up-to-date techniques to control not merely the tensile strength of NPK Socket Screws, but the *grain-flow* of the metal. In the illustration above, this grain-flow is most evident round the precision socket — where the utmost strength is needed.

At every stage in the production of NPK Socket Screws — from steelworks to stores — GKN Quality Control ensures uniformity, dimensional accuracy and good finish. NPK Socket Screws are made to engineers' standards — the highest. They are as good as they look.

NPK hexagon socket cap, set, countersunk and shoulder screws are available in a wide variety of sizes.
WEDGLOK self-locking hexagon socket screws and NPK hexagon wrenches are also available.

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Screw Division: Box 24, Heath Street, Birmingham 18. Telephone: Smethwick 1441. Telex 33-239

S/S 2310

The most powerful surface grinder in its class!

NEW ATLAS COPCO LSS82 Grinder — a multi-purpose machine

Surface grinding operations with flared cup wheel.

Cutting of a water pipe with high speed cut-off wheel.

Cleaning with wire brush.

High speed cleaning with hub wheel.

This expertly designed, heavy duty surface grinder, with a high-efficiency vane-type air motor, gives you greater power than any other surface grinder in its class. An exceptionally high air flow efficiency has given this model a 35% power increase over its predecessor. Yet for all its power it weighs only 11 lbs.

The new grinder is extremely versatile. Fitted with flared cup wheels, it can be used for cleaning castings and other surface grinding operations; with cut-off wheels attached, it can cut all types of metal, ceramics and stone! Wire brushes can be fixed to clean furnace scale, rust or paint from all large surfaces, and many high-speed cleaning jobs can be done by fitting hub wheels.

DESIGN FEATURES

Highly efficient vane-type air motor with cylinder, rotor and end plates precision worked from special steel alloys to give maximum life and efficiency.

Patented speed governor of a new design which holds speed constant at all loads and gives complete protection from wheel breakage due to excessive speeds.

Safety throttle of entirely new patent construction completely eliminates leakage.

Efficient exhaust silencer, integral with the motor housing, reduces exhaust noise to a comfortable level.

A complete range of compressed air equipment

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Atlas Copco AB, Stockholm 1, Sweden, or Atlas Copco (Great Britain) Limited, Maylands Avenue, Hemel Hempstead, Herts.

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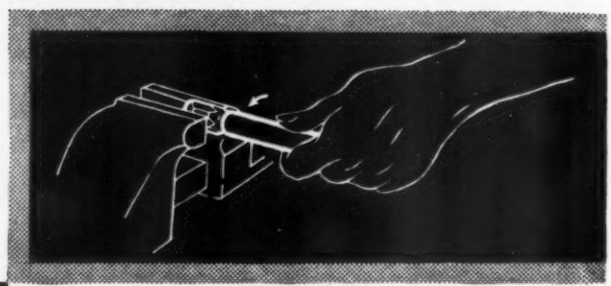
CURED IN SIXTY SECONDS

You can be your own doctor when it comes to providing the remedy for a broken or damaged hose-line. The obvious prescription will be AEROQUIP, with its detachable and re-usable end fittings — so easy to fit and so economical to use. If you specify AEROQUIP at birth your brainchild will be relieved of many teething troubles.



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ROTARY PUMP SEALS **ROMET** WITH MONUM FACES

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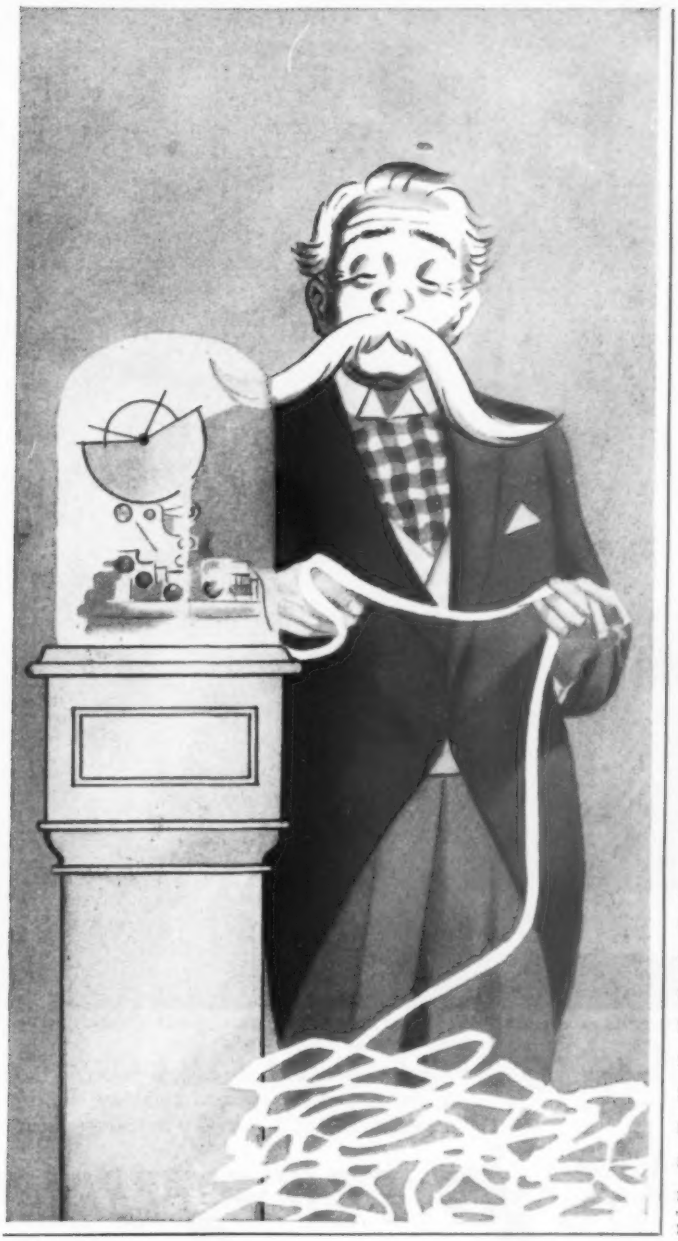

QUICK!

According to a Broker's mum, the City yesterday was at fever pitch.

Desoutters were up. Gilt Horses were stable.

And the Used Sock Company had changed hands 14 times in as many minutes.

A carefully-spread rumour that Desoutters were worth a fortune in unused soap coupons began a frenzied spate of buying. First in the queue was the M.D. who remarked that the whole affair smelt very fishy indeed. "Must be these hosiery fellows," he said as they changed hands for the fifteenth time.

Saying of the Week

"I was not aware that any other power tool manufacturers had made a bid for Desoutter. I was not aware that there were any other power tool manufacturers!"

— Managing Director of Desoutter

(Advertisement)

Desoutter puts power into your hands

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CRC 320

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The MORRIS

MINI-MINOR

and the

AUSTIN 7

have a most interesting braking system described overleaf and evolved by

Lockheed

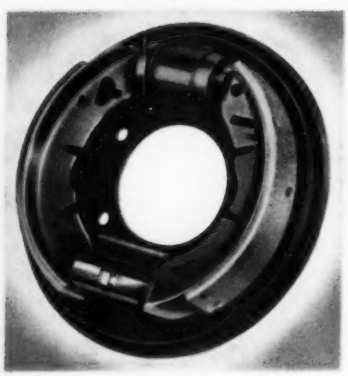
REGD. TRADE MARK

ONE OF THE AUTOMOTIVE PRODUCTS GROUP

THE MORRIS MINI-MINOR

...how Lockheed meet the of this extremely

The cars are lively, they have generous passenger space and may be expected to be well-laden—braking must be generous in order to match the performance.



A front brake

SPECIAL PROBLEMS IN BRAKING BALANCE

These new small cars have a high ratio of loaded to unloaded weight and this presents special problems in maintaining a correct balance of braking under all conditions. Lockheed solve the problems with their new pressure limiting valve which is most effective in controlling the power of the rear brakes during sudden stops.

Pressure limiting valve

BRAKING SYSTEMS FOR EVERY

ONE OF THE AUTOMOTIVE PRODUCTS GROUP

AND THE AUSTIN SEVEN 850

braking requirements interesting design

Lockheed drum brakes having leading and trailing shoes are used for both front and rear, giving a stable all-round performance and well-balanced braking.

This valve, it will be remembered, is also used on the Wolseley 6-99 and Austin A-99 cars fitted with Lockheed disc brakes.

HAND BRAKE ACTUATION

Weatherproof and neat, well-suited to the independent suspension, has the brake rod lying fore and aft with a floating lever reacting directly against both shoes. Double back-plates shield the drums against wet and dirt. A duplicate Lockheed master cylinder and a slave unit, are used for the clutch actuation.

Master cylinder

A rear brake

TYPE OF VEHICLE

Lockheed

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In the MORRIS MINI-MINOR and the AUSTIN 7 the high performance engine is protected by the Purolator 'MICRONIC' filter— leader in effective filtration

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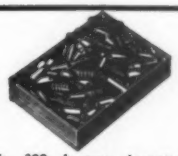
Just the job for you experimental people—a simply unlimited assortment from our tremendous range of springs of every variety. The boxes shown here are only a few examples—why not let us send you our illustrated list showing them *all?*

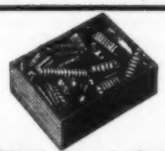
No. 760. 3 doz. Assorted Light Compression Springs. 1" to 4" long, 22 to 18 S.W.G., ¼" to ½" diam. 6/6 each.



No. 388. ½ gross Assorted Small Expansion Springs. ½" to 1½" long, ¼" to ½" diam., 18G to 21G. 9/6 each.

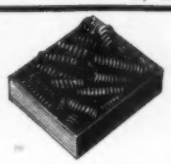
No. 757. Extra Light Compression, 1 gross Assorted, ⅛" to ⅜" diam., ½" to 2½" long, 27 to 19 S.W.G. 15/- each.

No. 466. ½ gross Assorted Small Expansion Springs ½" to 1½" long, 3/32" to 3/16" diam., 21G to 24G. 6/6 each.



No. 758. Fine Expansion Springs. 1 gross Assorted ½" to ⅞" diam., ½" to 2" long, 27 to 20 S.W.G. 15/- each.

No. 753. 3 doz. Assorted Light Expansion ¼" to ½" diam., 2" to 6" long, 22 to 18 S.W.G. 10/6 each.



No. 98A. 3 doz. Assorted 1" to 4" long, ½" to ⅞" diam., 19G to 15G. 5/6 each.

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Lighting - 1

Good lighting affects productivity in three ways, (1) by directly increasing the speed of working and reducing errors and wastage, (2) by improving overall factory efficiency through better supervision and housekeeping, (3) by providing better working conditions, thus improving labour relations and avoiding frequent changes of workpeople.

It is not possible to judge by the eye alone whether the lighting in any factory is good enough to serve these purposes because the eye has a tremendous range of adaptation (vision of some kind is possible within an illumination range of 1,000,000 to 1) and is hence an unreliable measuring instrument. Severe mental and eye strain or unconscious slackening of working speed may occur under lighting which *appears* to be adequate.

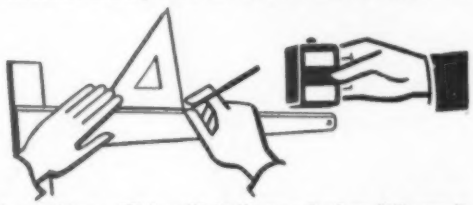
The only reliable way to appraise factory lighting is to conduct a lighting survey using a lightmeter, an inexpensive instrument which measures the actual illumination available. These figures can then be compared with official recommended values for the various tasks.

Lighting Survey

A methodically conducted lighting survey is the first step towards achieving good productive lighting. The average artificial illumination at working height should first be measured. This requires a number of readings at various positions relative to the lighting fittings, particularly beneath and between fittings and by the wall. The average of the readings should then be compared with the illumination values recommended by the Illuminating Engineering Society. The following is a general guide to illumination requirements.

NATURE OF WORK, PROCESS OR MATERIAL	ILLUMINATION LUMENS/SQ.FT.
Rough or routine work. Large detail. Medium to light material of good contrast.	7
As above, but work rather more skilled or critical.	10
Ordinary work usually involving workers' inspection. Medium detail and contrast.	15
Fairly critical work, fairly small detail or poor contrast.	20
Skilled work, small detail or dark material.	30
Fine or critical work, very small detail, very poor contrast or very dark material.	50
Very fine exacting work.	100
Minute work.	200

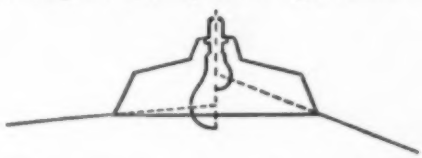
In addition, readings should be taken at selected working points with the lightmeter so placed that it measures the light on the work while the operative



is working. This will indicate whether full use is being made of the light or whether shadow is obscuring part of it.

Glare

Any direct light on the eye tends to reduce its sensitivity—thus reducing the power to see. Increasing the intensity of the light source may



therefore not improve matters unless care is taken to avoid glare either by correct positioning or by the use of correctly matched reflectors and lamps.

Walls and Ceilings

Certain surfaces and certain colours absorb light and therefore do not make the best of a light source; others reflect light and, so to speak, feed back on to the work a portion of the lighting which would otherwise be lost. Bright walls, moreover, have a good psychological effect which makes for contented —and therefore productive—operatives.

Shadows

Unnecessary shadows may seriously slow down work and also cause accidents. Lightmeter readings should always be taken under conditions exactly similar to those obtaining while work is going on. The operative himself may mask his work, an overhead crane or a heavily loaded conveyor belt may periodically obscure a light fitting, or a dust-laden atmosphere may reduce the designed lighting values.

For further information, get in touch with your Electricity Board or write direct to the Electrical Development Association, 2 Savoy Hill, London, W.C.2. Telephone : TEMple Bar 9434.

Excellent reference books are available on electricity and productivity (8/6 each or 9/- post free)—"Lighting in Industry" is an example.

E.D.A. also have available on free loan in the United Kingdom a series of films on the industrial uses of electricity including one on industrial lighting. Ask for a catalogue.

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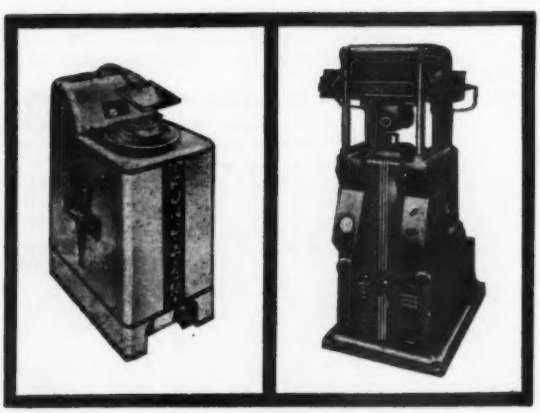

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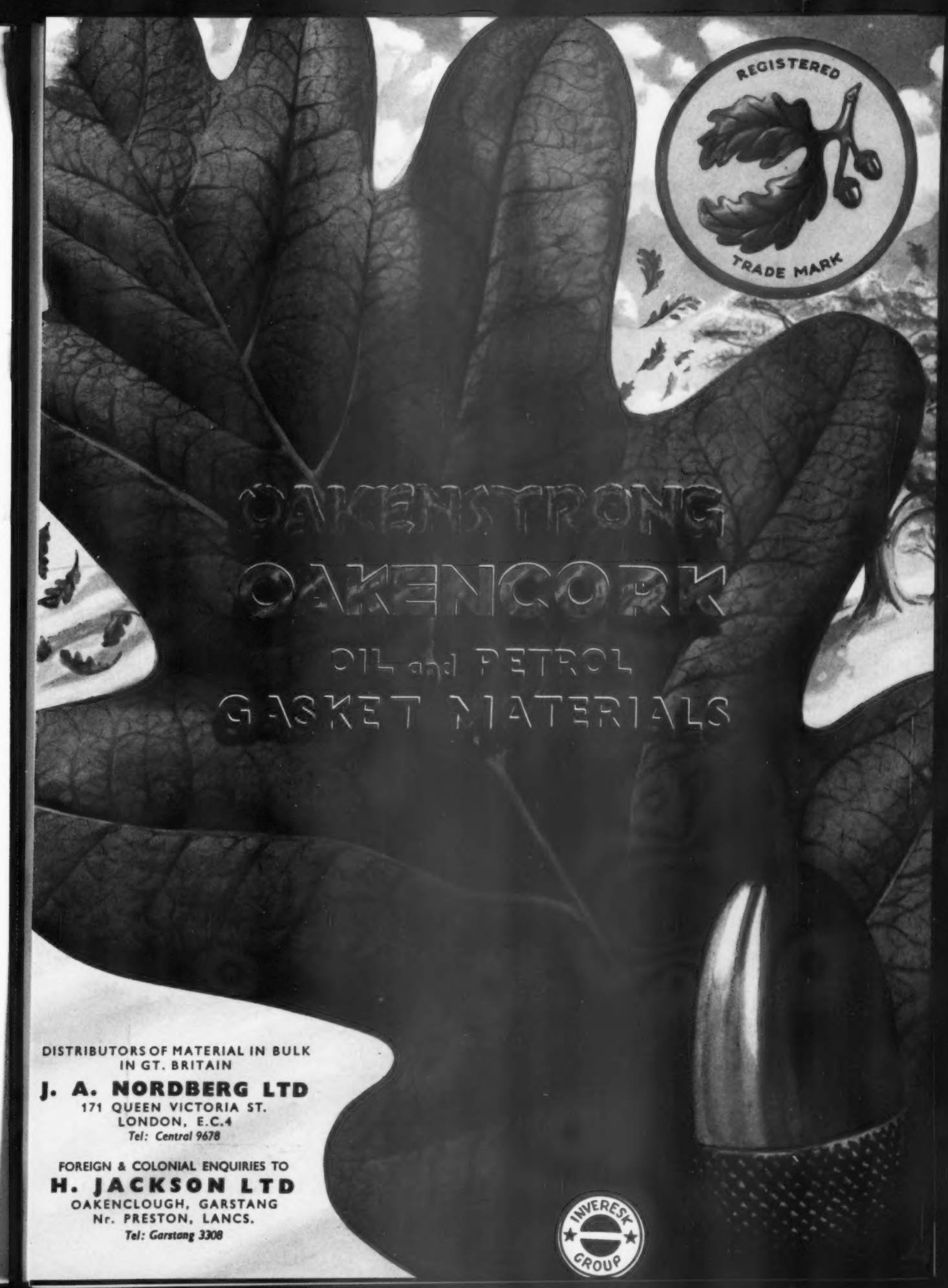

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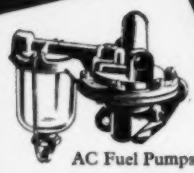
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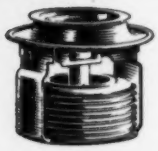
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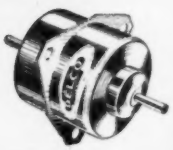
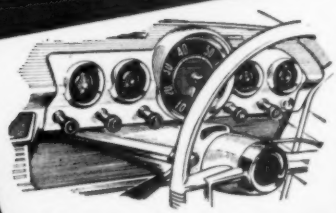

DELCO Electric Motors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a good vehicle starts with

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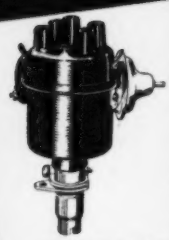
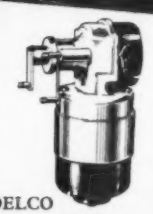
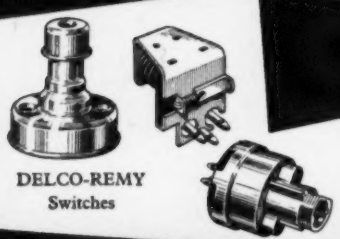
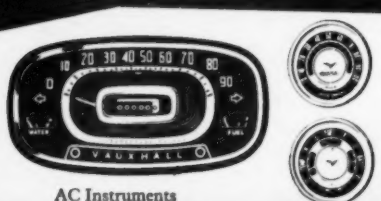
AC Oil Filters


DELCO-REMY Ignition Distributors


DELCO Electric Screen Wipers


DELCO-REMY Switches



AC Instruments

DELCO-REMY Electric Horns

Tests of Quality

Major Events—January to August 1959

DATE	MEETING	RESULT
Jan. 10th	New Zealand Auckland Grand Prix	Cooper-Climax 1st, 2nd, 3rd.
Jan. 25th	Monte Carlo Rally : Charles Faroux Trophy : Class Award over 2,000 cc. : Coup des Dames : R.A.C. Challenge Trophy : Class Award up to 1,000 cc.	Jaguar 3-car team. Jaguar 1st & 2nd. Austin A.40 1st. Austin A.40 1st. Austin A.40 2nd.
Feb. 7th	New Zealand. Teretonga Race	Cooper-Climax 1st & 3rd.
Feb. 8th	Canada: Winter Rally, Ladies Cup	Austin A.40.
Feb. 26th	Italy: Sestriere Rally: Touring Cars 500-1,000 cc. Touring cars over 1,600 cc.	Austin-Healey Sprite 1st. Jaguar 3·4 1st.
Mar. 15th	France, Lyon-Charbonieres Rally: Ladies Cup	Austin A.40.
Mar. 21st	U.S.A. Sebring: Touring Cars Class 4	Austin-Healey Sprite 1st, 2nd & 3rd.
Mar. 30th	Ireland International Rally: Ladies Trophy	Morris Minor 1000.
Mar. 30th	Goodwood : Lavant Cup : Sussex Trophy : Production Saloon Car Race	Cooper-Climax 1st. Lister-Jaguar, 1st & 2nd. Jaguar 3·4 1st, 2nd & 3rd
Mar. 30th	Mallory Park : Sports Cars over 1,200 cc. : Formula Libre	Lister-Jaguar 3·4, 1st. Lister-Jaguar 3·4, 1st.
Apr. 11th	Oulton Park : British Empire Trophy : Sports cars over 1,500 cc.	Cooper-Climax, 2nd. Cooper-Monaco, 1st. Lotus-Climax, 3rd. Lister-Jaguar, 4th.
Apr. 18th	Aintree : Sports Cars over 1,100 cc. : 2,000 cc. Class : Saloon Car Race : Scratch Race Saloon Cars 1,600-2,600 cc.	Lotus-Climax 2nd. Lister-Jaguar, 3rd. Lotus-Climax, 1st. Cooper-Climax, 2nd. Jaguar 3·4, 1st, 2nd & 3rd. Jaguar 3·4, 2nd.
Apr. 26th	Italy, Syracuse Grand Prix: Formula 2	Cooper-Climax 3rd.
May 2nd	Holland, Tulip Rally, General Classification Class F: over 2,600 cc	Jaguar 3·4, 1st. Jaguar 3·4, 1st & 2nd.
May 2nd	Silverstone: International Trophy, Formula 2 Sports Cars, 2,000-3,000 cc. Sports Cars, 2 litre Sports Cars, 1 litre Production Touring Cars, General Class Production Touring Cars, over 2,600 cc. International Touring Cars, 1,300-1,600 cc.	Cooper-Climax, 1st & 3rd. Lister-Jaguar, 2nd. Cooper-Climax 1st. Lotus-Climax, 2nd & 3rd. Jaguar 3·4, 1st, 2nd & 3rd. Jaguar 3·4, 1st, 2nd & 3rd. M.G. 2nd.
May 10th	Monaco Grand Prix	Cooper-Climax, 1st & 5th.
May 16th	Pau Grand Prix: Formula 2	Cooper-Climax, 1st, 2nd & 4th.
May 24th	Targa Florio: 1,100 cc. Sports Class	Austin-Healey Sprite 3rd.
May 31st	Dutch Grand Prix	Cooper-Climax, 2nd & 3rd. Lotus-Climax, 4th
July 5th	French Grand Prix, Rheims	Cooper-Climax 3rd.
July 18th	British Grand Prix, Aintree Sports cars over 2 litres Sports cars 1,500 cc. to 2 litres	Cooper-Climax, 1st & 3rd. Lister-Jaguar, 2nd, 3rd & 4th. Lotus-Climax, 1st & 3rd. Cooper-Climax 2nd.
Aug. 23rd	Portuguese Grand Prix (Subject to official confirmation.)	Cooper-Climax 1st & 2nd.

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THE BRITISH PISTON RING CO. LTD., COVENTRY, ENGLAND

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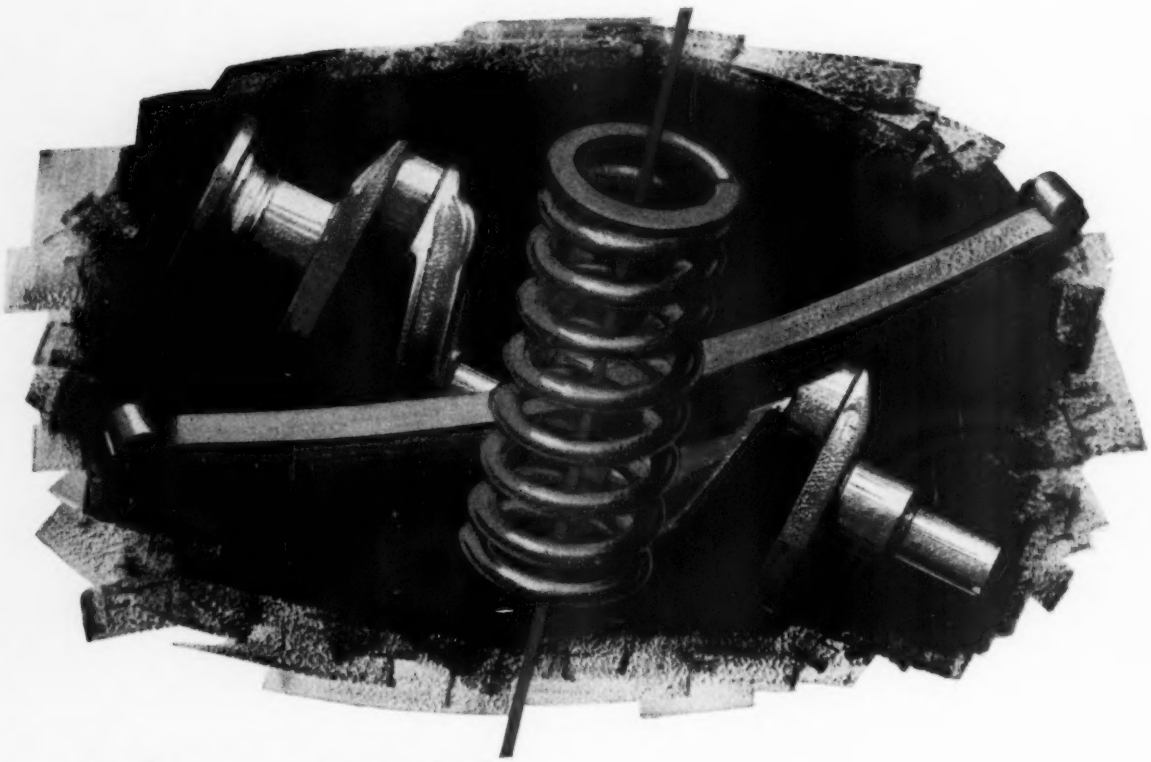
When you reflect upon MODERN COMFORTS—



think of R·T·B

The decoration at the top of the page is a reminder that RTB activities, so deeply associated with sheet steel and tinplate, also include the production of the alloy steels used for vehicle springs, crankshafts and other vital parts.

But for our pioneering of the continuous-strip mill in this country, there would not be enough steel sheets to meet the present scale of car production.

Our contribution to modern comforts does not stop there, for our products are to be found in every kind of modern household equipment—in streamlined cookers and refrigerators, in easily-cleaned stainless steel, in radio sets, electric cleaners—even in electric radiators.

In fact, there is hardly anything you do—anywhere you go—without enjoying the unseen help of RTB.

Richard Thomas & Baldwins Ltd.

R·T·B

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helps smooth out problems

it is the continued research of Midcyl that helps smooth the way of the Auto Engineer with such of his problems as are associated with Cylinder Blocks, Cylinder Heads, Camshafts and Brake Drums

MIDCYL

THE MIDLAND MOTOR CYLINDER CO. LTD., SMETHWICK, STAFFS

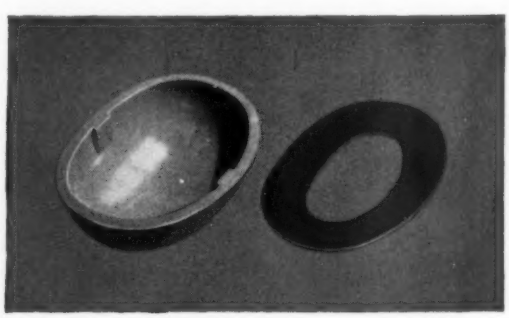
H-W.79

Here's a typical example of how the mechanical strength of Beetle D.M.C. (Dough Moulding Compound) helps to solve industrial problems.

These ear defenders are used to protect the hearing of workers in proximity to jet engines and in noisy factories. During the development work by the makers, Dennis Ferranti Meters Ltd., twelve sets moulded from conventional materials were delivered to Bomber Command for testing. The report of The Central Medical Establishment, Acoustics Laboratory, was that the ear defenders gave excellent protection, but that the mouldings then used were considered too fragile.

D.M.C. mouldings were substituted and have been found virtually indestructible.

D.M.C. is the ideal material whenever high mechanical and/or electrical strength is required. The material can be moulded in conventional compression presses on short cure times and at low pressure. We shall be glad to send full technical details.



This illustration shows the main moulding from Beetle D.M.C. and a retaining piece made from Scarab material. The Earguards are used in engine testing, rivetting, blasting, rolling mills, weaving sheds, paper mills and on aerodromes.

B·I·P

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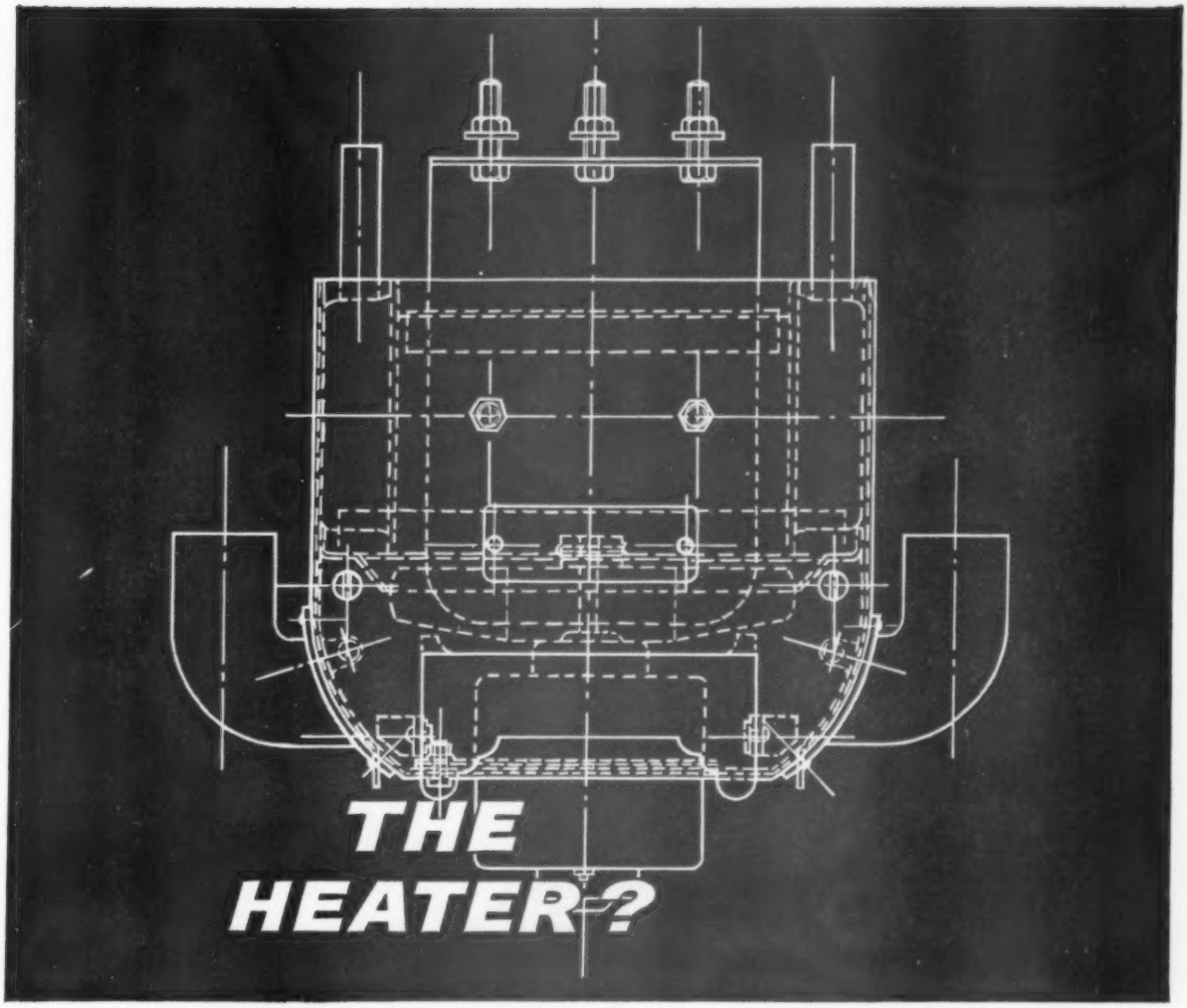
B.I.P. CHEMICALS LTD., Oldbury, Birmingham. Telephone : Broadwell 2061.
London Office: 28, Haymarket, S.W.1. Telephone: Trafalgar 3121



Let's talk to Delaney Gallay!

Delaney Gallay have a very great deal of experience in the design and development of car heating, demisting and air conditioning equipment, together with extensive production capacity.

Whilst you are still in the design stage, why not have a chat with them? They will be glad to design and produce something to suit your requirements.

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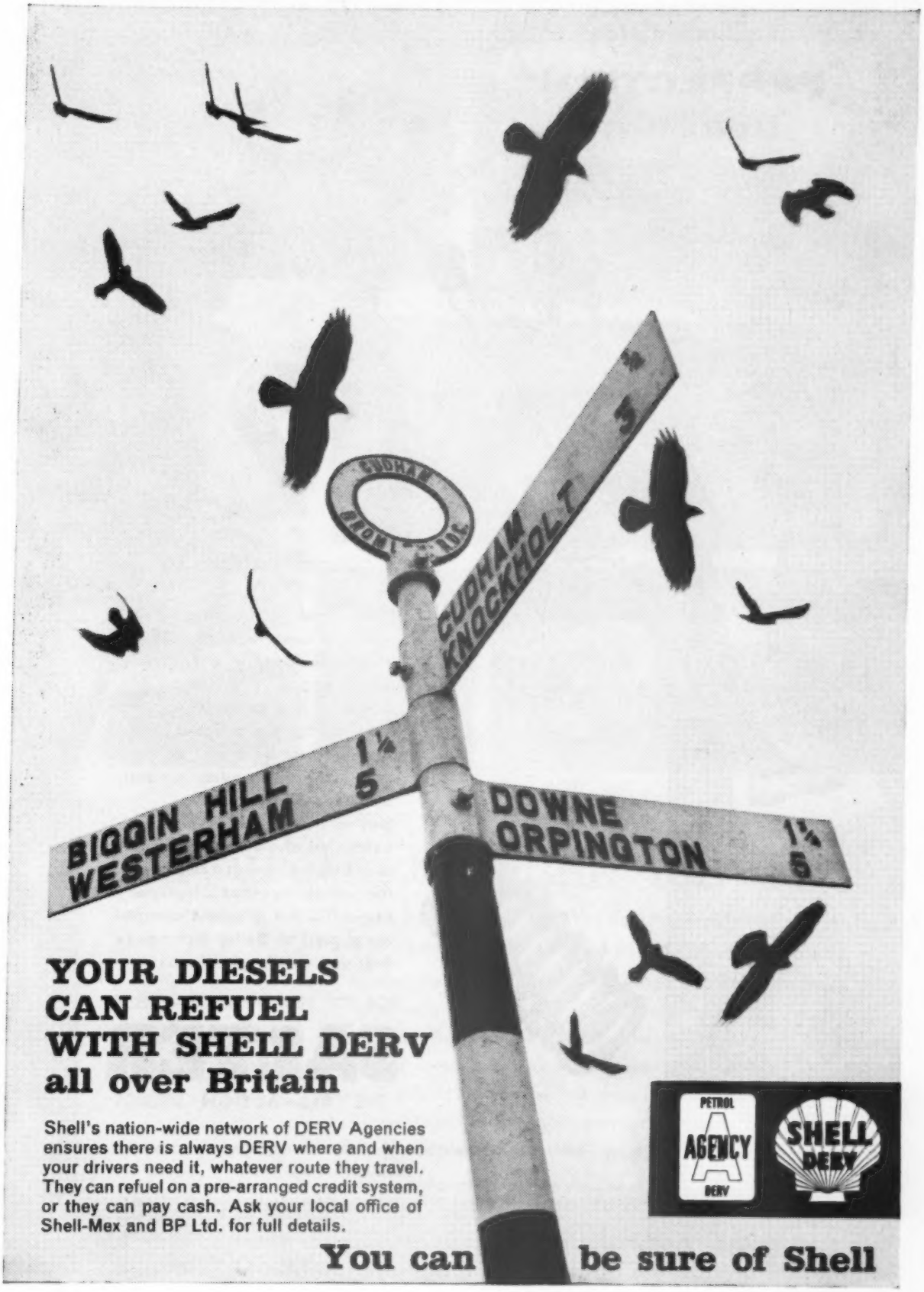
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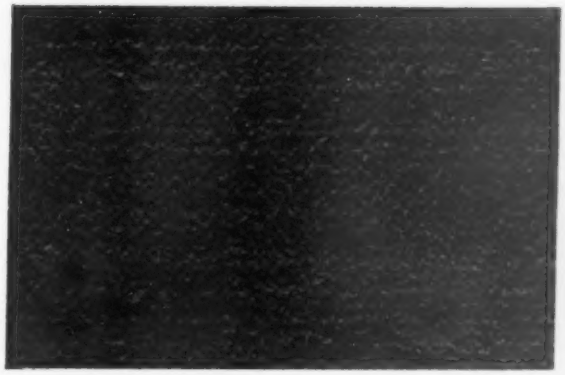

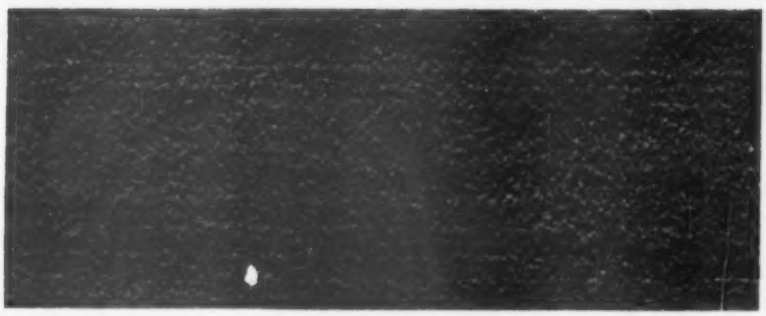

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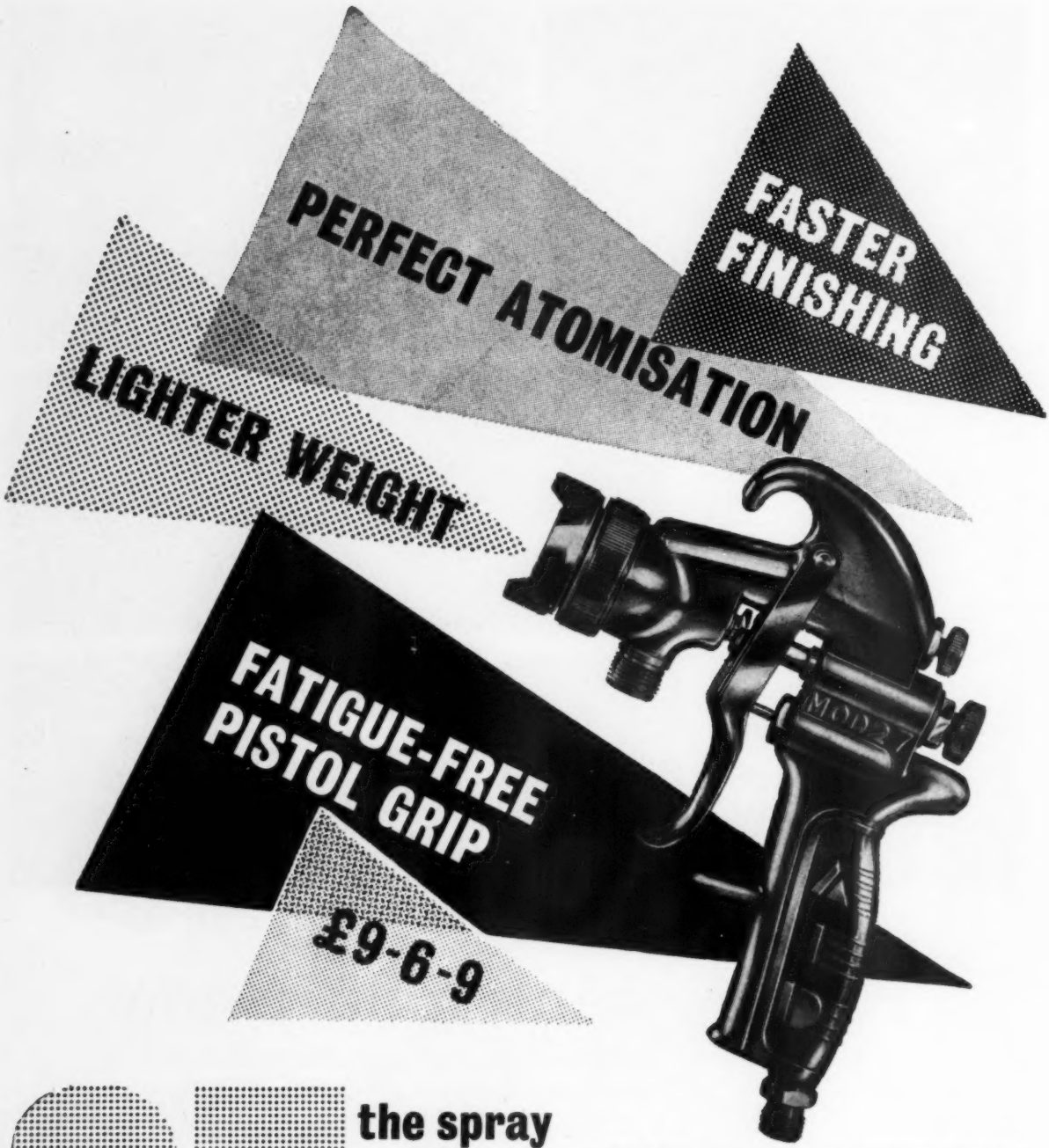

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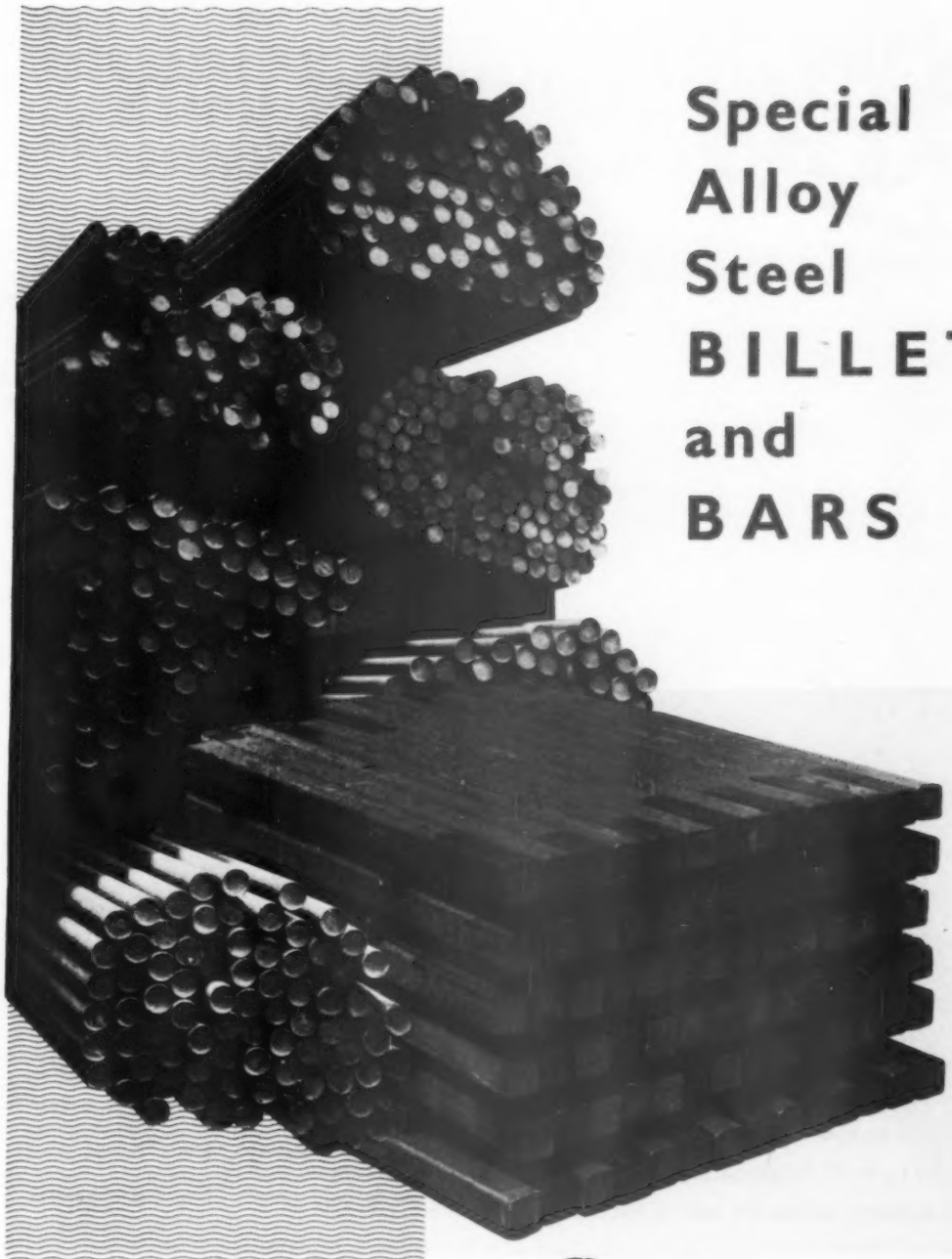

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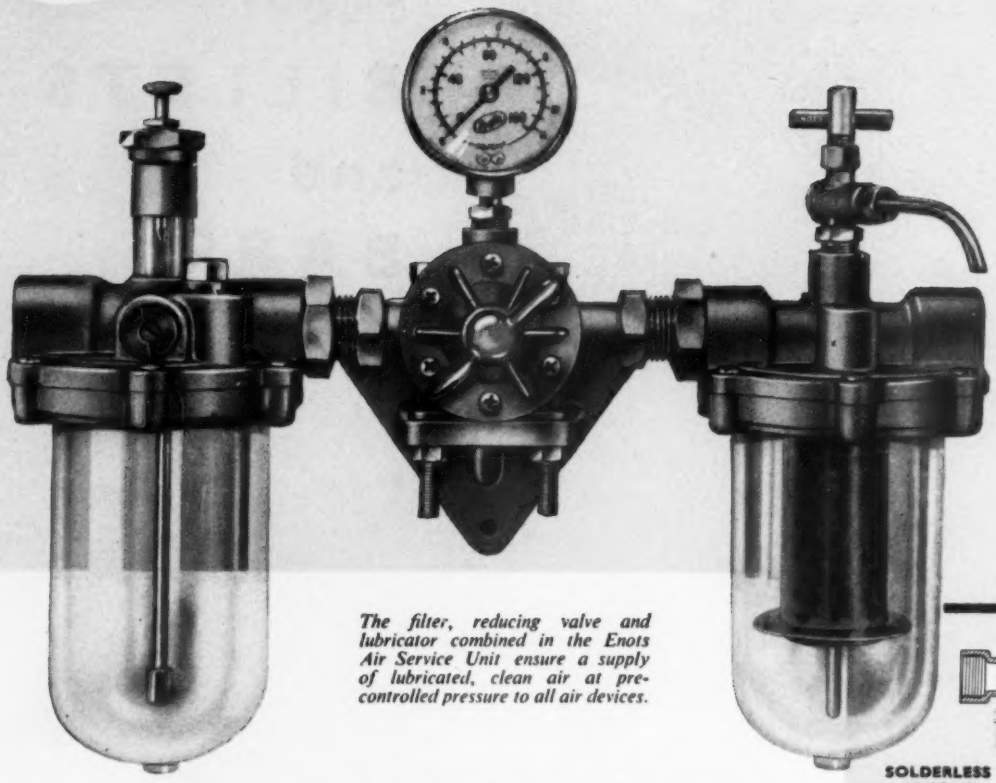
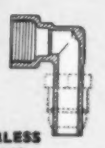


The filter, reducing valve and lubricator combined in the Enots Air Service Unit ensure a supply of lubricated, clean air at pre-controlled pressure to all air devices.



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A pressure reducing valve fitted to almost any pneumatic device will save its cost in a very short time by reducing air consumption. At how many points in your works are you using full line pressures of up to 100 lb. per sq. inch where 80, 60 or even lower pressures would be adequate? Savings are more or less in direct proportion to pressure reduction.

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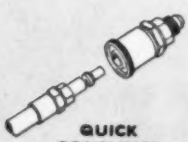

QUICK COUPLINGS

EJECTOR VALVES

LEAFLET AS357 describing our Air Service Units and individual components and other catalogues on Enots pneumatic devices are available on request.

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Cleaning component parts for the new Hoover Junior Cleaner

At the highly mechanised Hoover Factory at Perivale in Middlesex a Dawson metal cleaning and degreasing machine plays an important role in the efficient production line for the new Hoover Junior cleaner. The illustration above shows how the three diecast components converge ready for washing. The two smaller components come from the bottom left and the housing comes down the chute from the top left.

This is only one of the countless applications for the Dawson metal cleaning and de-greasing machines which are widely used in every grade of engineering from electric shaver components to heavy diesel cylinder blocks.

A catalogue giving full details of these machines can be sent, on application, from Drummond-Asquith (Sales) Limited at the address given below.

Machines for cleaning large and small articles

Dawson

DEGREASING AND CLEANING PLANT

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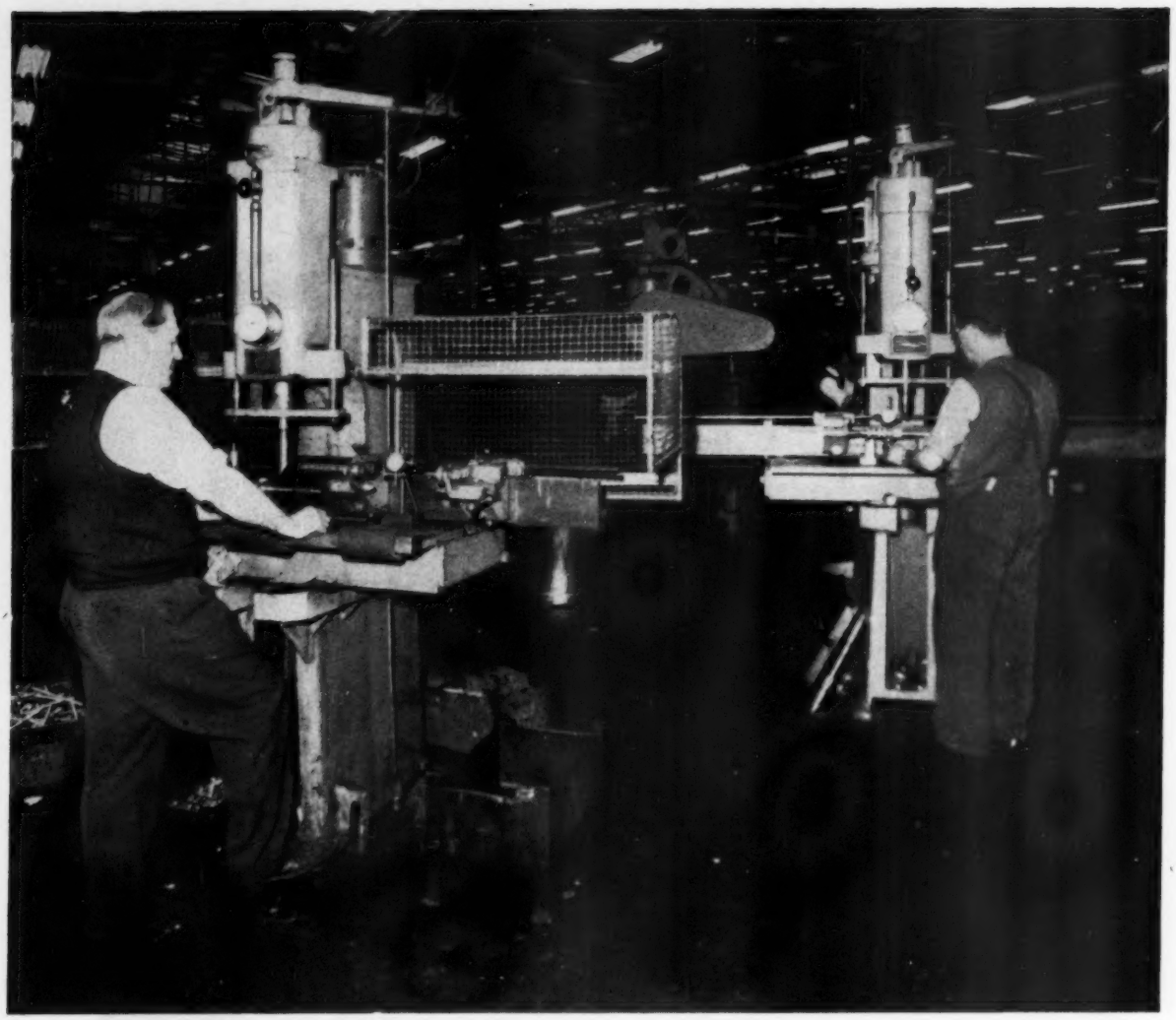
London Works, 486 Roding Lane South, Woodford Green, Essex. Tel: Wanstead 7777 (4 lines)



HI-TON

PRESSES at Vauxhall Motors Ltd.

Two Hi-Ton Hydraulic Presses are here shown truing-up gearbox mainshafts and main drive pinions preparatory to grinding.

These modern hydraulic presses may be considered the standard presses of the Automobile Industry for straightening, assembling, etc. Any pressure from zero to the maximum of the press can be applied and is predetermined by setting an adjustable pressure control valve. No excess pressure is necessary and maximum tool life is assured.

Hi-Ton Hydraulic Presses are available in a range from 1-300 tons

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HF 279

MAXICUT

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No. 2A
Gear Shaping Machines
with Automatic loading

These Drummond Gear Shapers can now be supplied with automatic loading equipment for the mass production of specific types of gears. When operated in conjunction with conveyor systems, gear production becomes a completely automatic process.

The illustration above shows a line of Maxicut machines installed for high output production of automobile gears and in the close-up view gear blanks can be seen stacked in the tubular type magazine. From this magazine a slider type feeder loads the blank on the work spindle and displaces the machined component into an exit chute. A push button panel at the left of the machine incorporates switch controls for hand or automatic operation, also controls to repeat or not-to-repeat the cycle. The indicator light panel on top of the machine shows whether all functions are operating correctly or not.

Write to-day for full details of Drummond Maxicut Gear Shapers for high speed accurate gear production.

This view shows the slider type feeder arrangement and magazine containing gear blanks.

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D331

QD... Q&A

Q What exactly do you mean by QD ?

D It's a piston for re-vitalising tired engines.

Q Tired ?

D After twenty or thirty thousand miles, especially on heavy duty work, most engines tire a little. Tiring is expensive. Your accounts are showing that several of your vehicles are heavy on oil. Your drivers are complaining of low power and are taking longer over their journeys. Fundamentally, that's largely due to cylinder wear.

Q In other words, you think I need new pistons ?

D Probably. It depends upon the amount of wear involved. It is possible that the trouble can be cured with Duaflex alone.

Q Duaflex ?

D The D part of QD. When piston wear is not excessive you can fit Duaflex Rings to existing pistons to compensate for cylinder wear, restore power, cut oil consumption, and give thousands more miles of lively performance. All for about a sixth of the cost of a replacement engine.

Q And if the pistons are badly worn ?

D Then new QD pistons are the answer, complete with Duaflex and Quickseat Top Rings. Your service engineer can determine the best course after measuring the bores and pistons. You can rely on these products curing the trouble. Wellworthy have been specialising in this field almost since motoring first began. **Q.E.D.**

If you have got vehicles in your fleet running up your lube oil bills, heavy on fuel and doing far too much low gear work, fit QD. Write now for literature and address of nearest Stockist to Dept. A9

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Air and Oil cooled MEDIA transformers	Low Voltage range 65 V	50-310 A
	High Voltage range 95 V	40-220 A
Oil cooled MASTA transformer	Low Voltage range 65 V	70-440 A
	High Voltage range 95 V	50-300 A
Air cooled MONTA transformer	Low Voltage range 50 V	30-200 A
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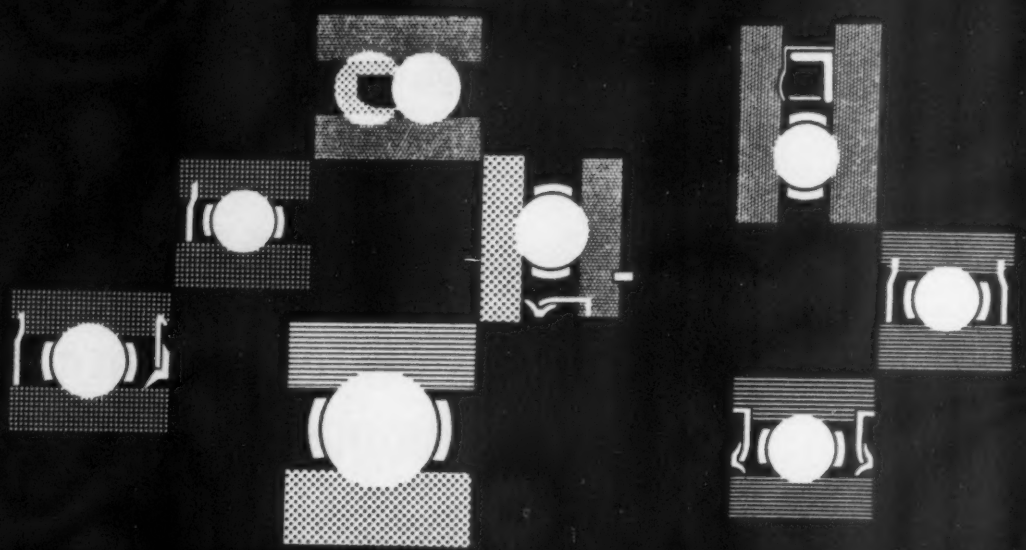

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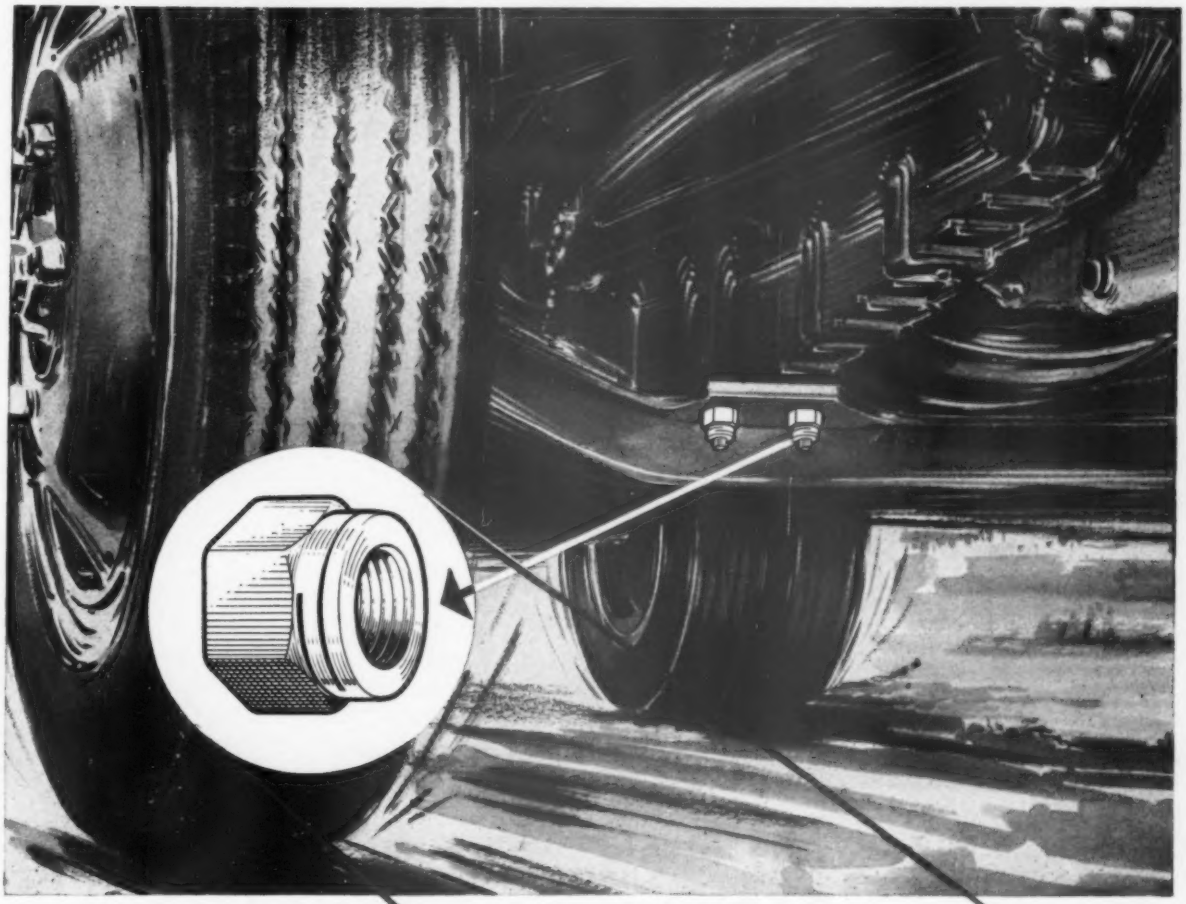

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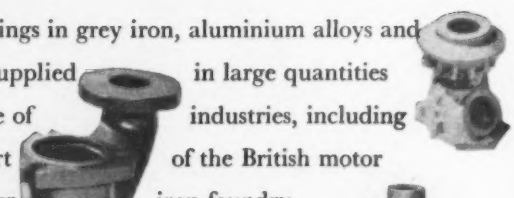

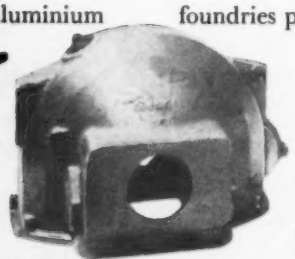

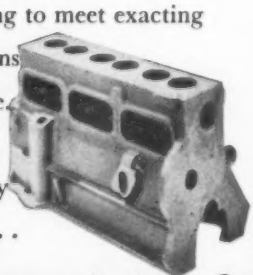

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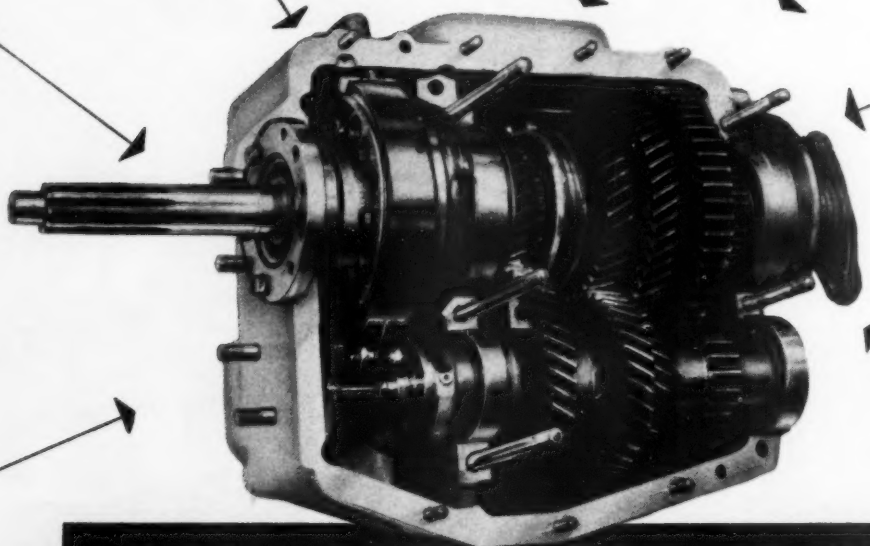

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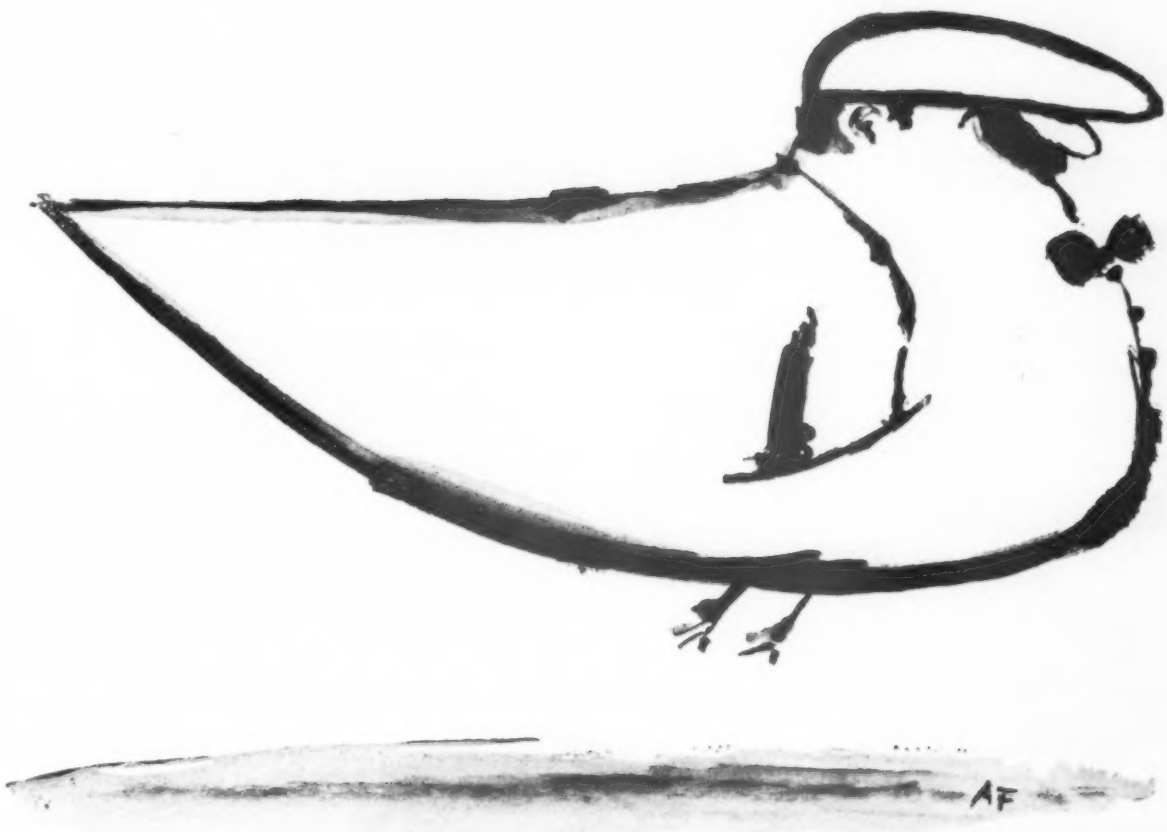

Time lost in youth, dreaming, dozing. Precious hours. Recapture must in stratosphere.
Shave London. Breakfast New York. Chase space.

Jettery, you board plane. Comfort. Calm. Barley sugar. Hostess mostest.
Five minutes . . . precious minutes. Breathe.

Wilmot Breeden where? Somewhere. Make seats? *No*. Make hostess? *No, no.*
Make glass of whisky? *No*. Well, where? What? Which? Tell.

Make inlet guide vanes! No? Fact? *Yes, yes, hollow blades, super-duper-special process.*
What do? *Guide air*. Where? *Out of sight, out of mind*. So!
Make anti-icing ducts, hydraulics . . . So! Fascinating.

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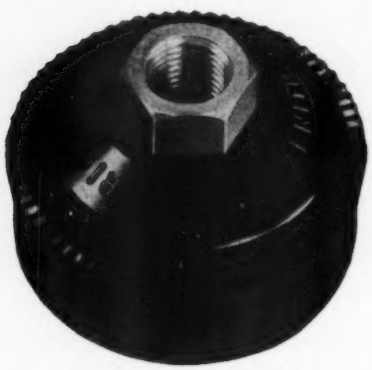

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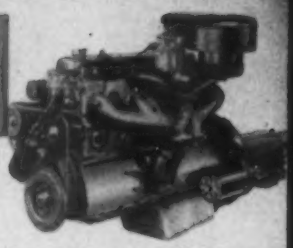
Extensive experience with so many different types of engine, coupled with modern test equipment, makes it possible for us to predict accurately the performance of Metalastik torsional vibration dampers. Moreover, by our design methods a damper can be designed so as to transfer critical frequencies to other and less critical ranges.

Lastly, Metalastik T.V. Dampers permit a great range of r.p.m. without detriment to the engine.

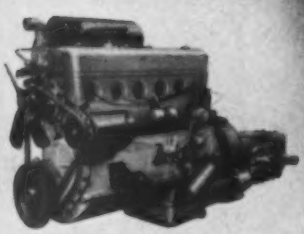
It is this unique combination of advantages that has made the Metalastik torsional vibration damper such a widespread success.

The small illustrations show some famous petrol engines with Metalastik T.V. Dampers. Equally successful on diesel engines, many world-known oil-engine manufacturers fit Metalastik dampers.

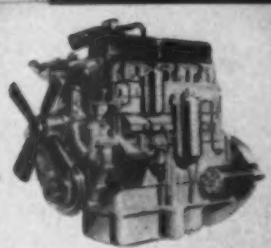
David Brown
Aston Martin 3-litre engine



B.M.C. 'C' series engine



Daimler D.K. 400 engine



Humber Hawk 2267 c.c. engine

Jaguar 3·4-litre engine

METALASTIK

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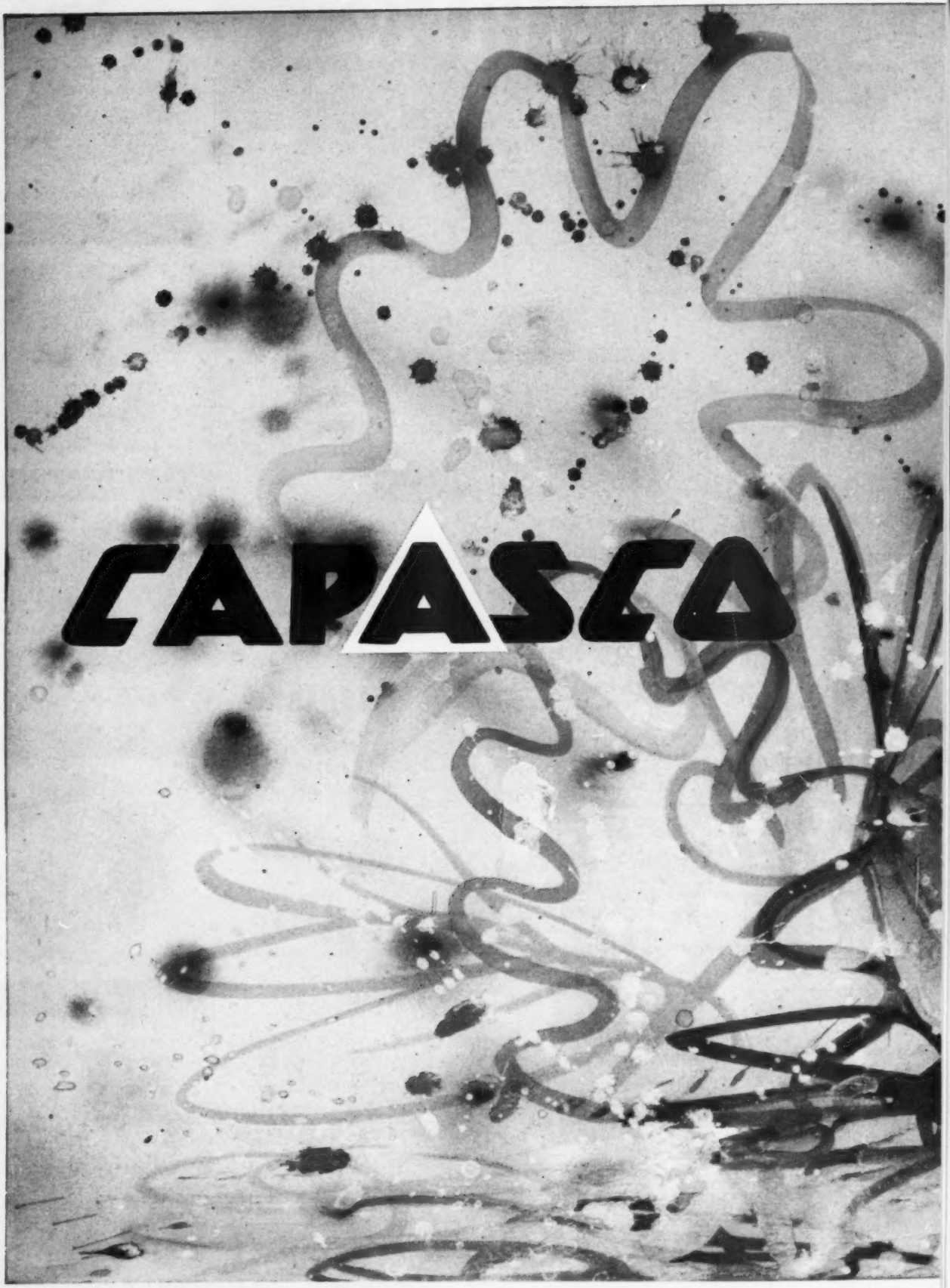
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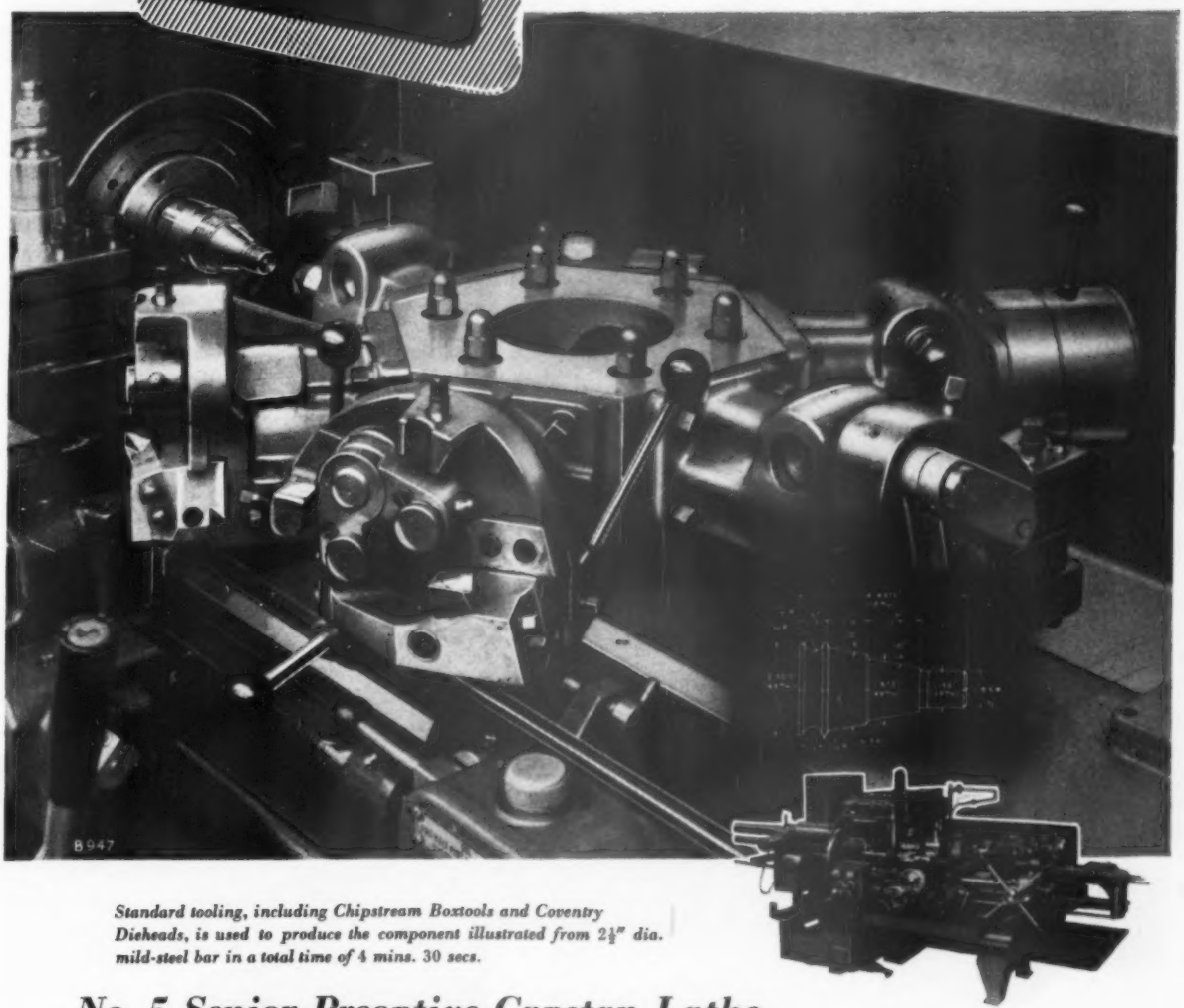

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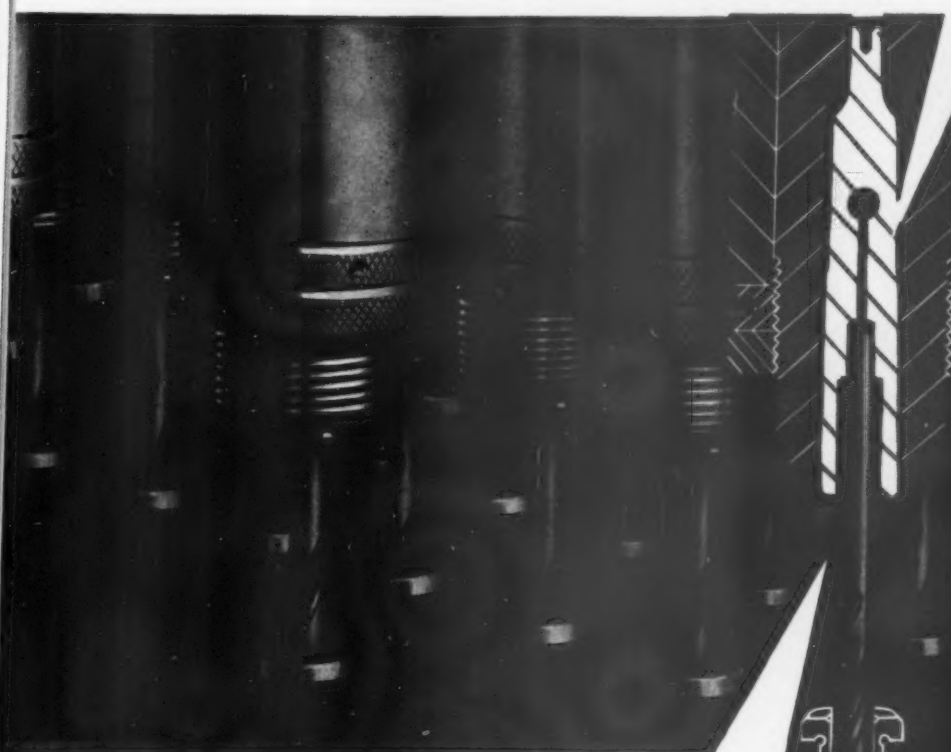

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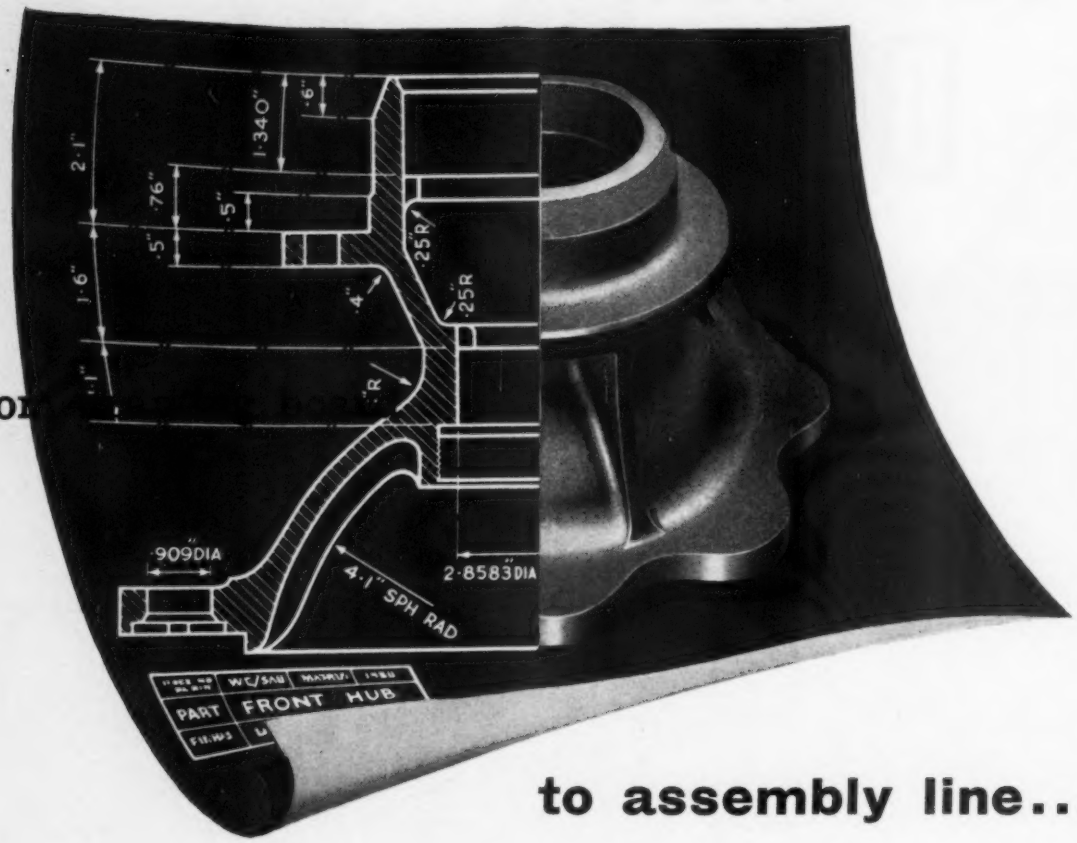

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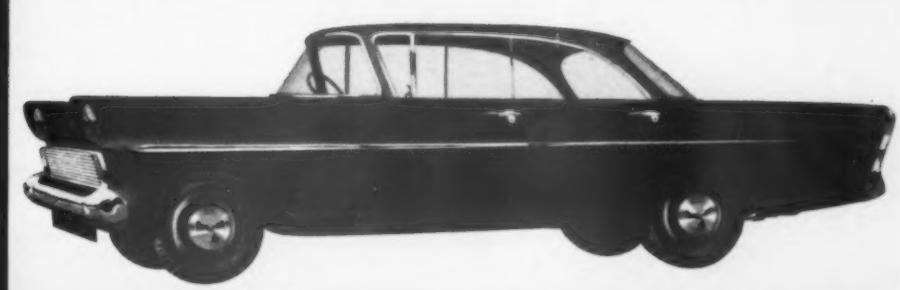

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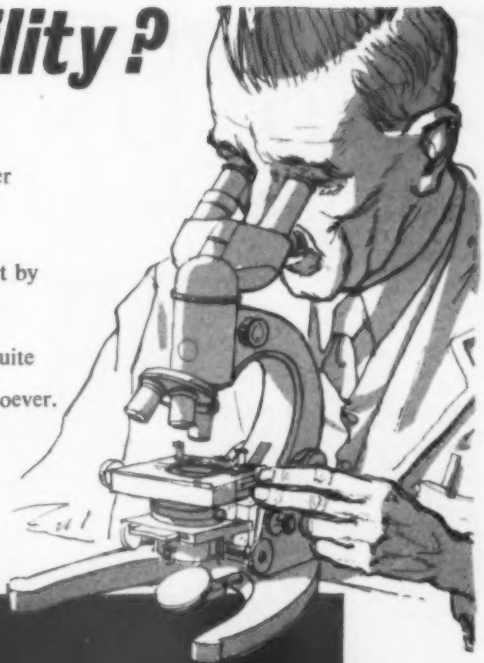
What is filter *Stability?*

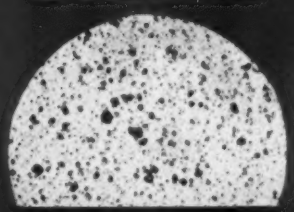


A filter should do its work with equal efficiency throughout its working life. It should be *stable*. The C.A.V. paper element fuel filter continues to work efficiently right up until it is finally choked.

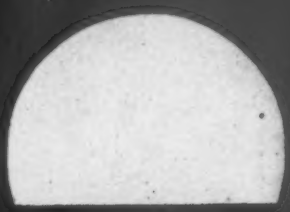
Many filters on the market do not, however. On test, they may start by showing a fair efficiency, but later, more and more of the abrasive particles pass through the filter material. Such unstable filters are quite useless, and are no safeguard to your fuel injection equipment whatsoever.

These photomicrographs of fuel tell the story of actual tests.

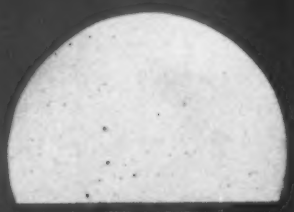
STABLE



1 Fuel laden with abrasive dust, before filtration

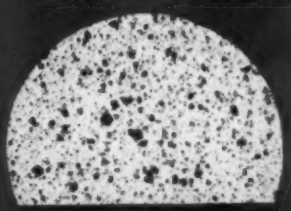


2 The same fuel after passing through a C.A.V. paper element filter at beginning of test

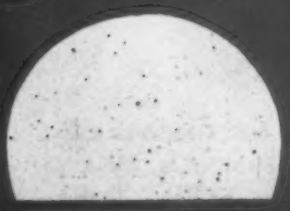


3 Fuel after filtration by the same element when all but choked by waxy sludge and dirt

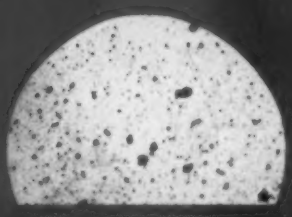
UNSTABLE



4 The same fuel is used for this test



5 At the beginning of test, filtration is fairly efficient



6 After the filter element is partly choked, the fuel passing through takes with it nearly all the particles of dust

There is only one genuine C.A.V. paper filter element. Substitutes are frequently unstable

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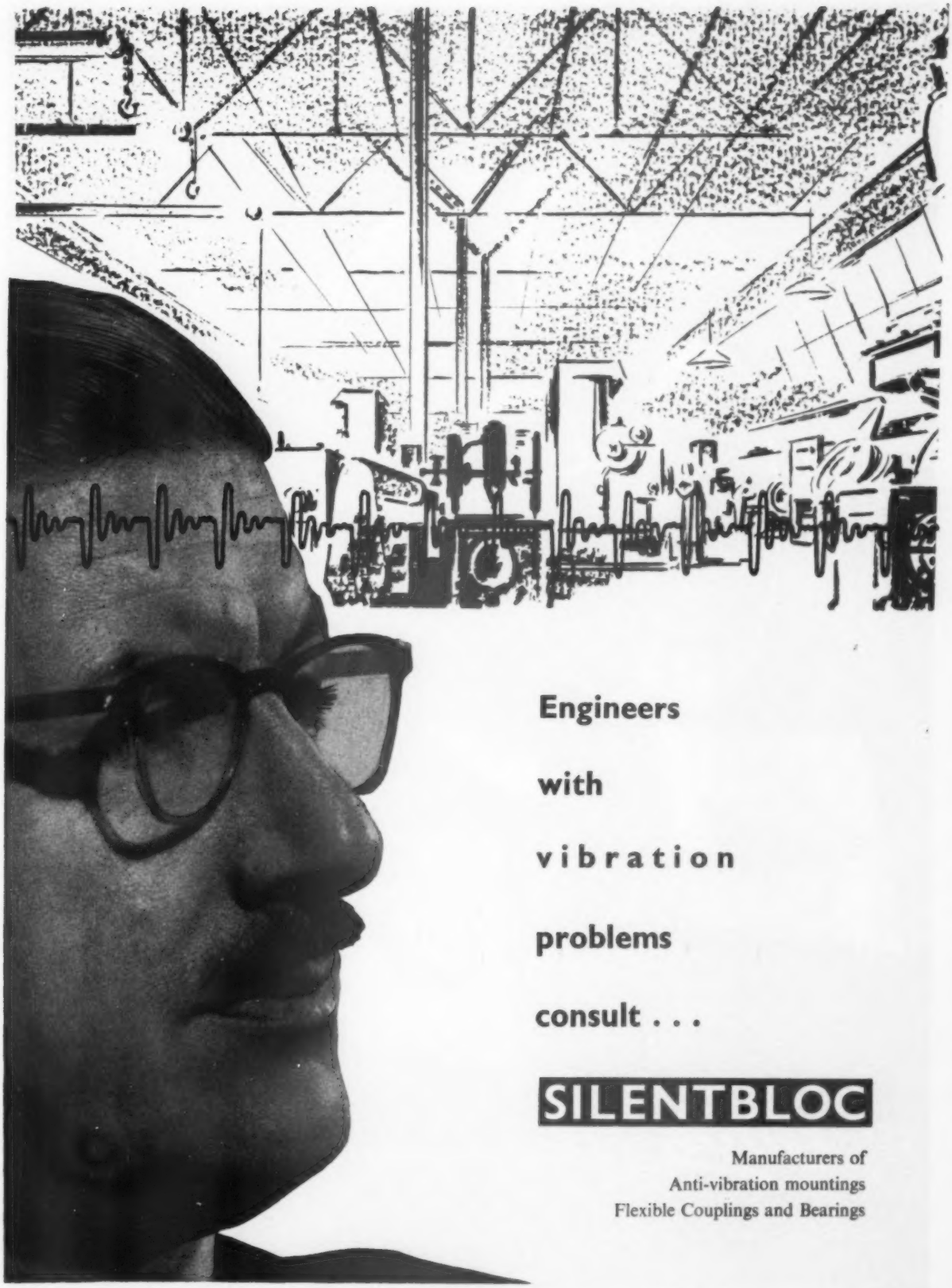

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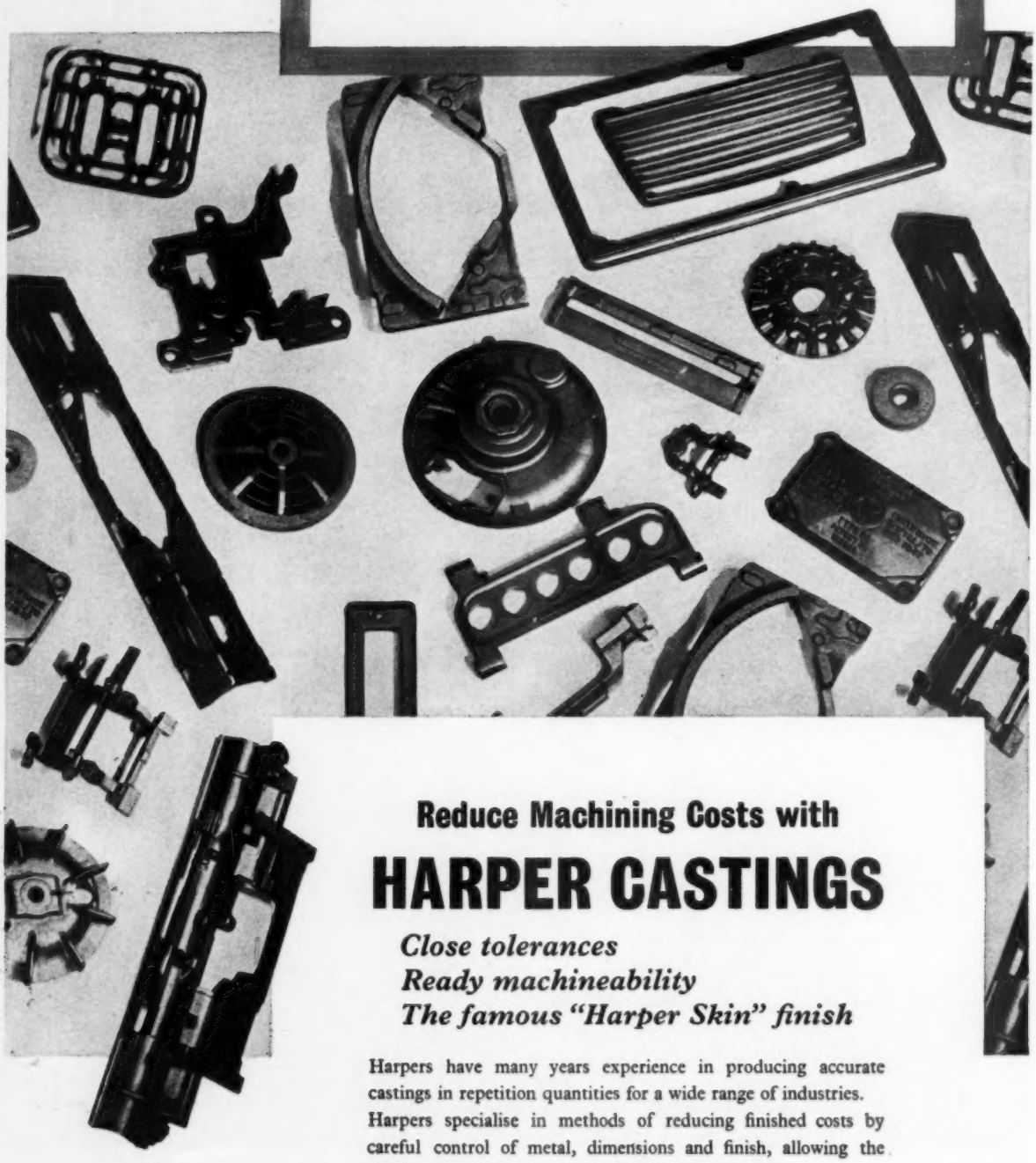

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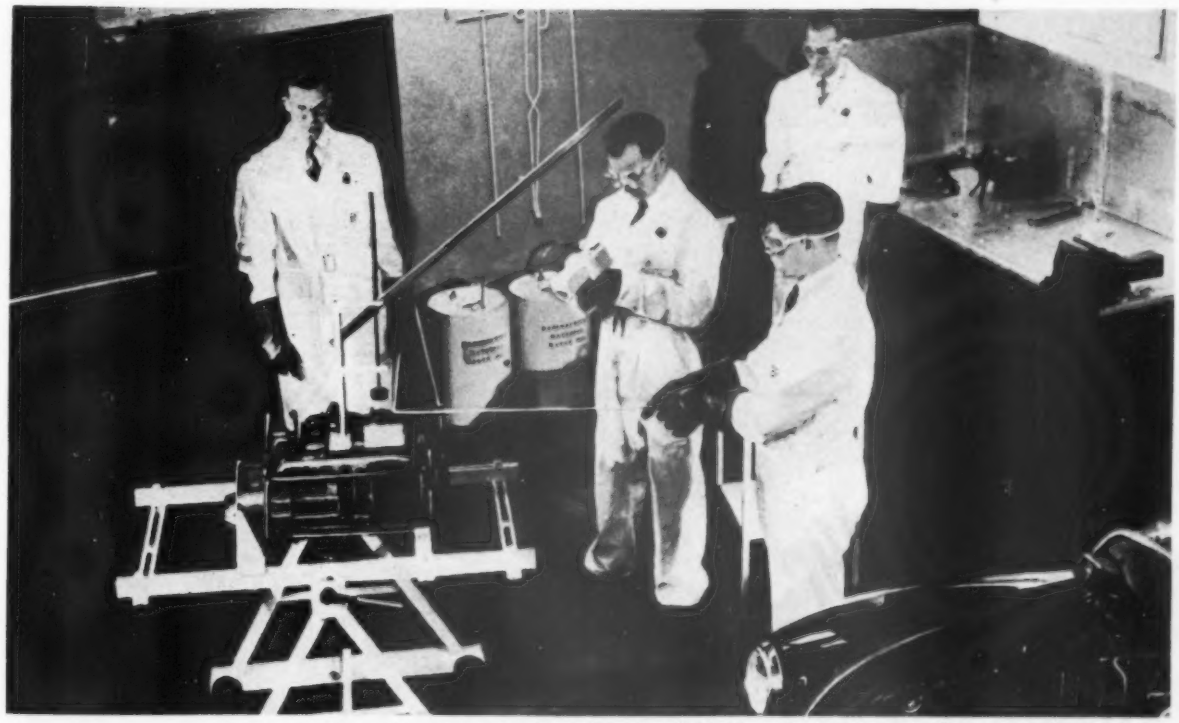

The radio-active piston ring is being installed in the engine in preparation for a test on the road. Many tests of BP 'Visco-static' with the radio-active wear detector proved 80% less wear on test piston rings.

Backed by radio-active wear detector and experience of motorists everywhere

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Tests with this oil showed 80% less wear on piston rings

TODAY YOU CAN BUY a motor oil with more certainty of what it will do than ever before. BP Energol ' Visco-static ' has been tested by the most modern of all methods – the radio-active wear detector. Results showed 80% less wear on test piston rings compared with ordinary oils.

Now motorists all over Britain are finding out for themselves the amazing wear reduction with BP ' Visco-static '.

How the test worked

Research scientists took an ordinary Morris car. They fitted it with a radio-active piston ring and they ran the car on sub-urban, town and country roads in Surrey. As the engine ran, infinitesimally minute fragments of metal were worn off the moving parts – including the radio-active piston ring – and these particles passed into the engine oil.

The 'clucking hen'

At the end of each day's running a sample of oil was taken and its radio-activity was assessed by a special Geiger counter – the famous ' clucking hen ' of atomic research. This allows the amount of metal worn from the piston ring during the day to be measured with amazing accuracy.

After a long programme of testing against the finest ordinary oils, the result showed 80% less wear with BP 'Visco-static'.

How BP 'Visco-static' saves wear

Most wear in engines occurs in the first few minutes after each cold start. That is with ordinary oils. When the engine is left overnight, acid products condense on the cylinder walls. These cause corrosive attack to take place. It is the corroded deposits which cause early wear when the engine is started again.

With BP ' Visco-static ' you fight this wear in two ways. Firstly, this advanced oil protects metal surfaces from acid corrosion when the engine is stationary. Secondly, BP ' Visco-static ' remains free flowing even in freezing cold so that every part of the engine gets proper lubrication from the moment the engine starts.

Saves petrol too

BP ' Visco-static ' is a different, more fluid kind of oil, and saves up to 12% in petrol because it cuts down oil drag. Starting is much easier too in all weathers.

BP 'Visco-static' is for use all the year round

BP ' Visco-static ' is for use all the year round in all 4-stroke car engines in good condition and for which an oil from the range SAE 10W to SAE 40 is normally recommended. There is never any need to change the oil because of change of season. You can use BP ' Visco-static ' all the year round, but you do of course have to change this at regular intervals.

To obtain maximum benefits of this outstanding oil you should have your old oil drained and make a complete change.

Ask for BP ' Visco-static ' – Britain's most advanced motor oil – at any garage where you see the BP Shield.

Rolls-Royce officially approve BP 'Visco-static' for all their post-war cars.

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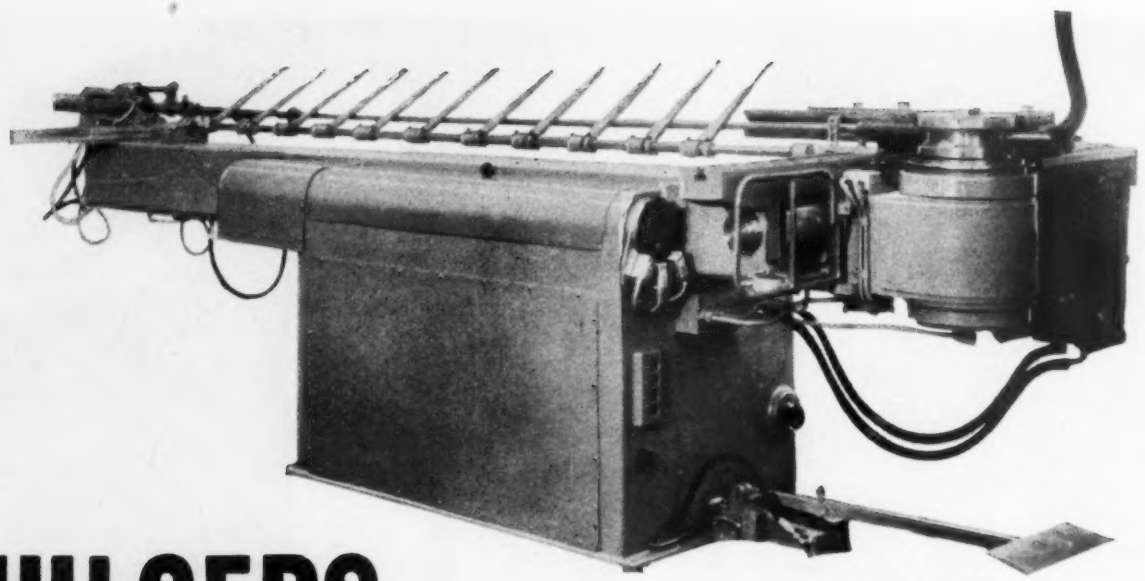

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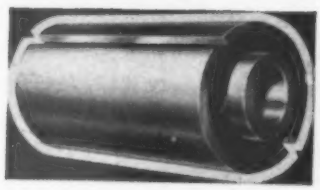

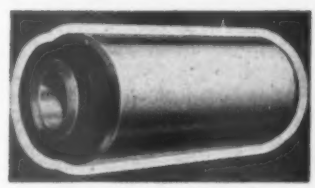

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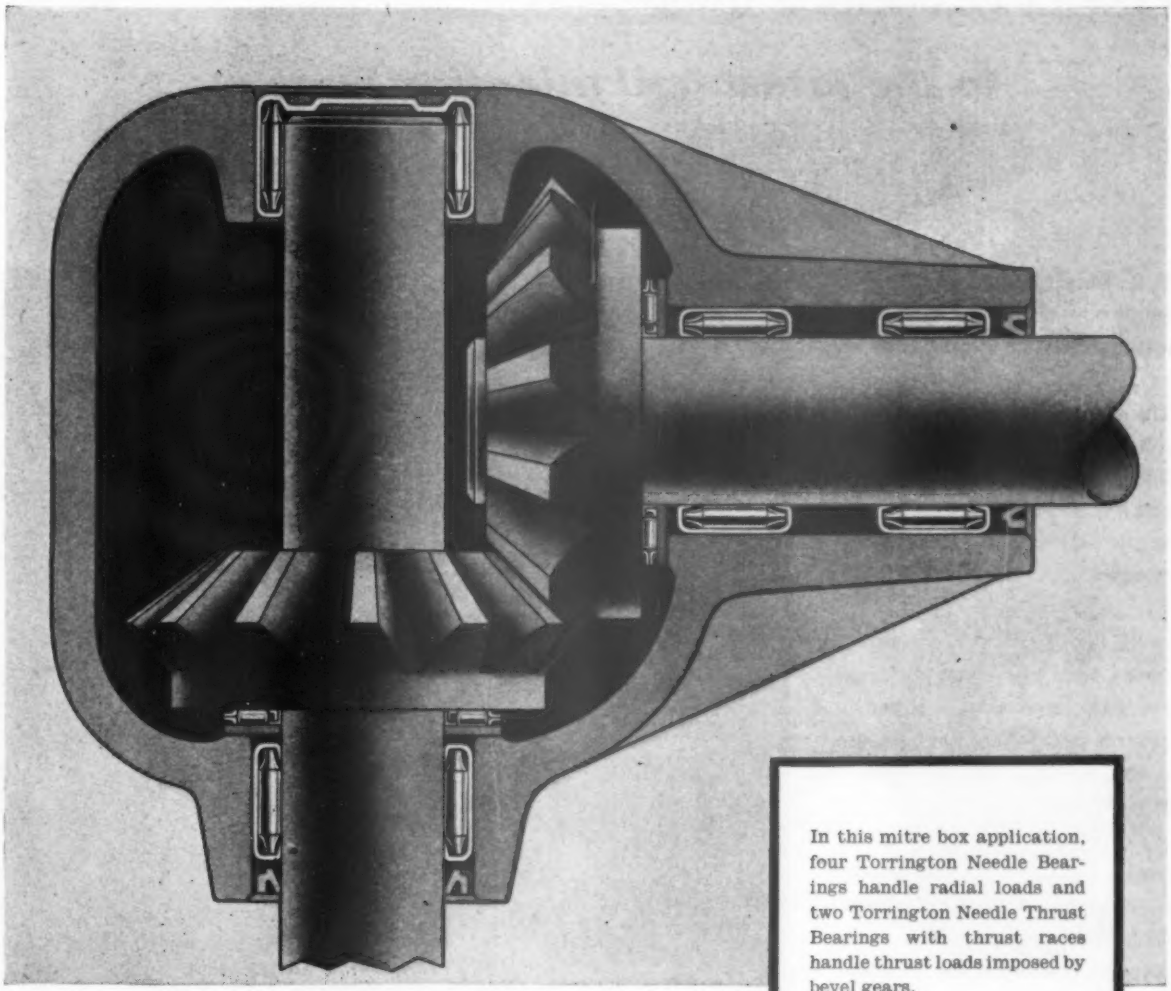

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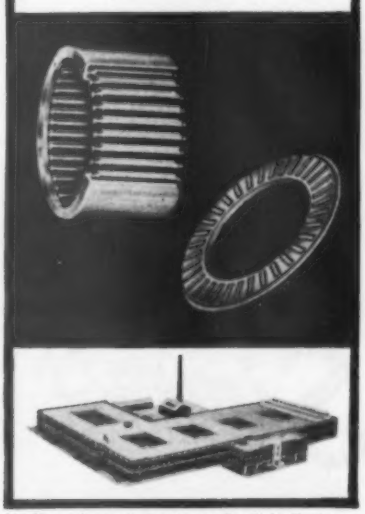

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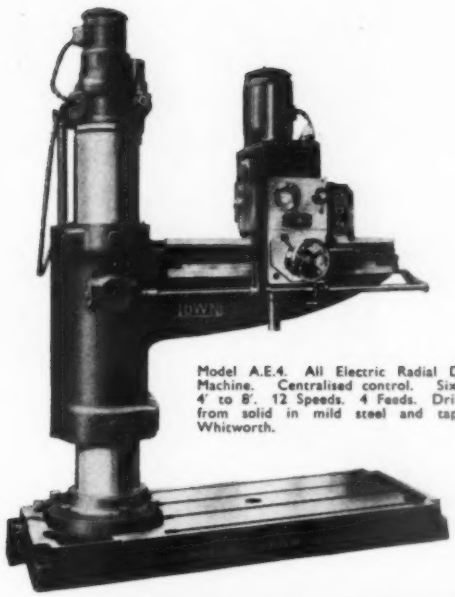

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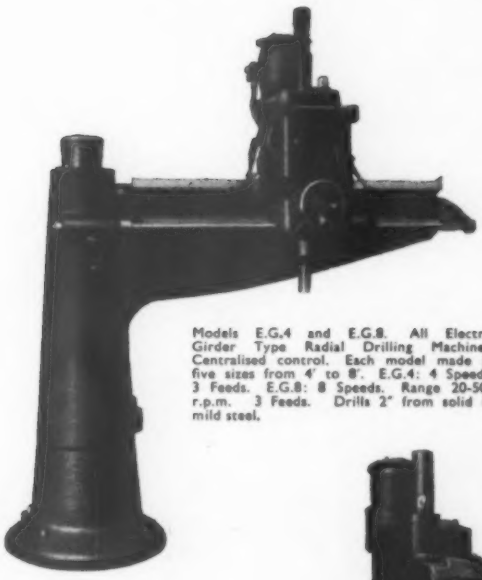

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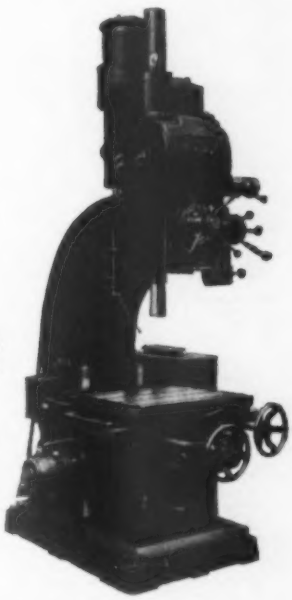

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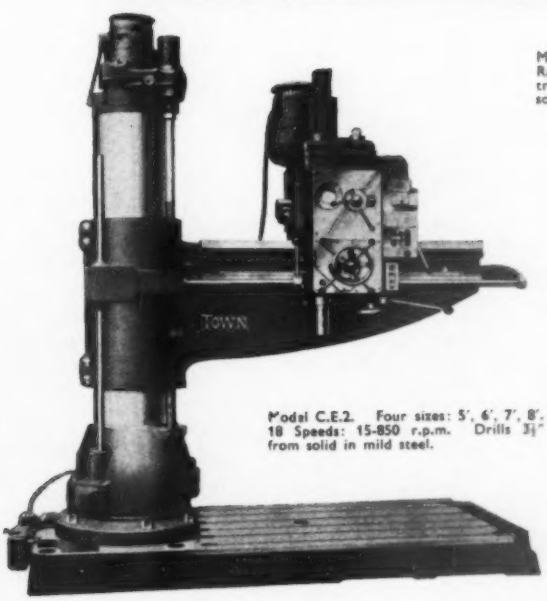

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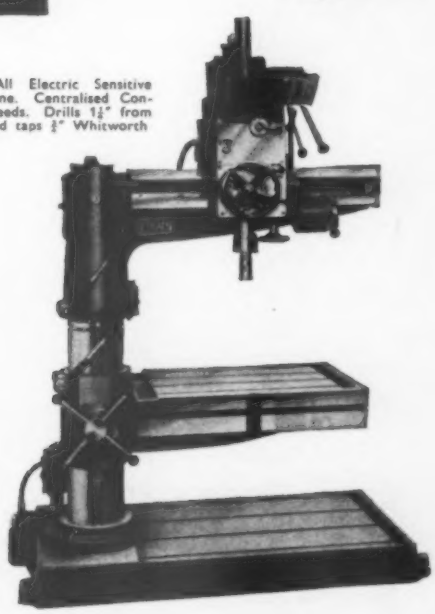

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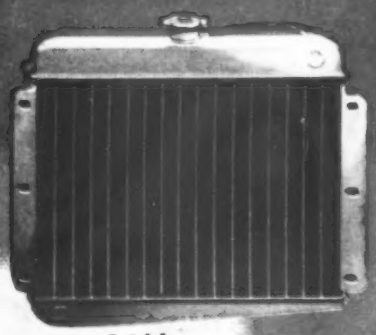

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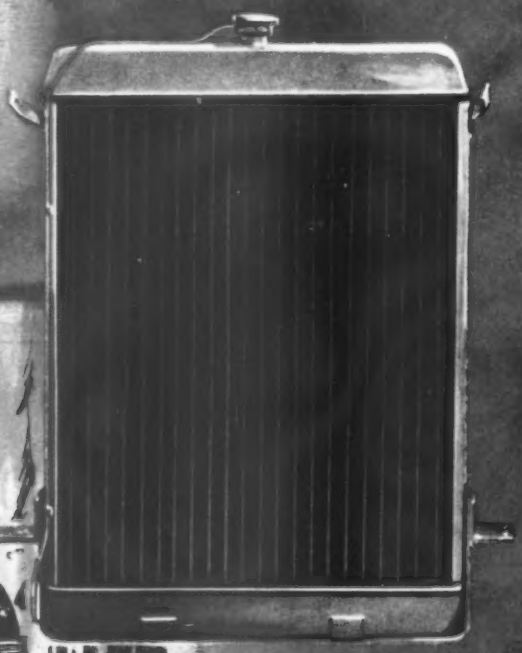

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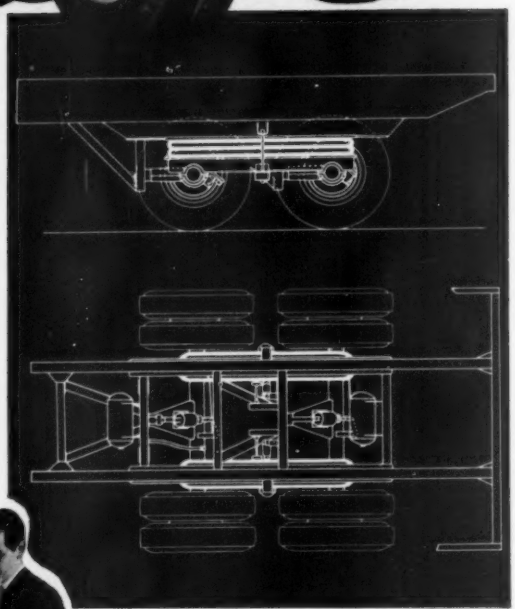

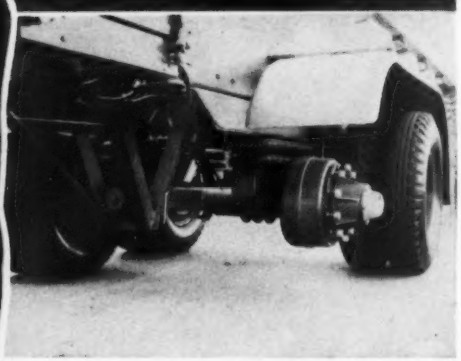

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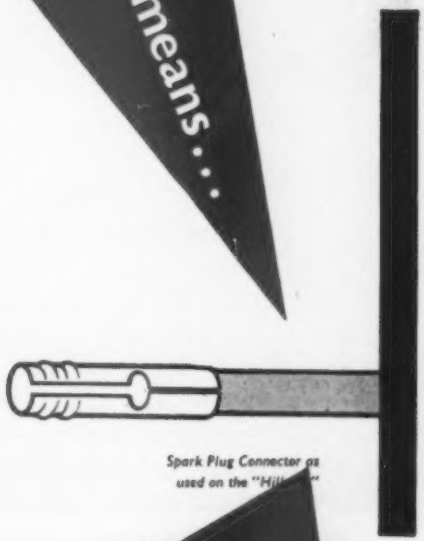

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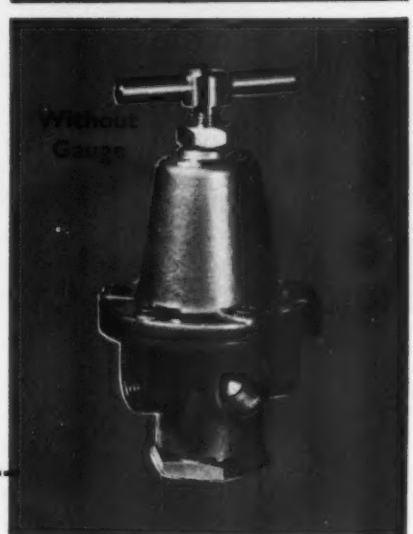

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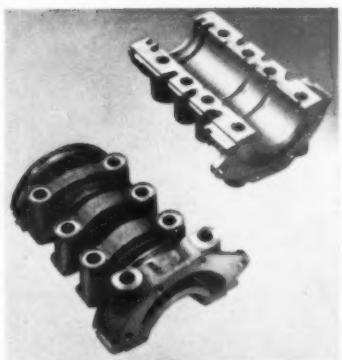

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Competition in America

SHORTLY, British manufacturers will have to face increased competition in the United States from locally produced small cars—in American terminology, a small car is, of course, approximately the size of European vehicles powered by engines of about 2 litres capacity. The question arises, therefore, as to what measures must be taken to ensure that vehicles produced in the United Kingdom are able to compete favourably. Perhaps the most difficult problem to be faced is that of producing vehicles at a cost which, when shipping charges are added, is still low enough. The number produced is, of course, the key to the problem, and in this respect British manufacturers have always been at a disadvantage. However, the demand for this type of car in the United States may not be high enough for the American firms to make their vehicles in the quantities to which they have hitherto been accustomed, but it must also be remembered that American cars of this size, if exported to Europe, will sell more readily there than the larger models that they have been producing up to now. Obviously, the recent trend in Britain, towards rationalization of models, will help, and ought to be extended.

To assess the potential market in the United States, it is necessary to analyse the various factors that have led to the demand for smaller cars there. One is, of course, traffic congestion, but there are others that are less obvious. Prices of new cars of the sizes currently produced in that country have been steadily increasing, and there is now a large demand for vehicles that are more economical in respect of not only first cost but also operating costs. Hitherto, the demand for less expensive cars has been satisfied by the second-hand market, but it is now thought that a high proportion of customers will prefer to buy a new small vehicle rather than a second-hand larger model at the same price.

With regard to operating costs, it is of interest that during its life-time a car will consume petrol which, in terms of monetary value, represents rather more than the first cost of the vehicle. Therefore, fuel consumption should not be ignored. Another important factor to be considered is that the American motorist generally does not expect to have to undertake preventive maintenance, the cost of which is relatively high: he demands inherent reliability, and has repairs and adjustments done only when the need for them becomes apparent. It cannot be emphasized too strongly that if our vehicles are to succeed in maintaining their position against the competition of local manufacturers, spares and other facilities must be immediately available on a nation-wide basis to a much greater extent than hitherto. Also, it is important to bear in mind the fact that, in such a large country, the efficiency of the selling organization is at least as important as that of the vehicle.

The next question to be considered is what measures, if any, should be taken to improve design. In general terms, the requirements of the American market are such that the vehicle must be able to cover very long distances without tiring its driver. Therefore, attention should be devoted primarily to reducing engine and other noises, and improving suspension. American motorists are accustomed to vehicles with low-periodicity suspension systems, but in this connection the principal difficulty is, obviously, that British manufacturers have to make cars that will be acceptable not only in America but also in the European and home markets, where very soft suspension is positively unpopular. The merits of independent rear suspension, as a means of obtaining a soft ride without reducing roll stiffness, should be re-examined.

Since the traffic congestion in American towns and cities is much worse than in Britain, acceleration is even more important there than here. British manufacturers have a considerable advantage, in the form of fully developed automatic transmissions suitable for cars of relatively small size. Although American motorists are reputed to like novelty, the foreign car is in itself a novelty to them, so it does not need to be equipped with all the gadgets that are so common on the vehicles produced recently in the United States.

When a new model is designed in Europe, its cost can, of course, be greatly reduced by the incorporation of components already in production for other models. In this respect American manufacturers will initially be at a disadvantage. However, there can be little doubt that European manufacturers will not win a significant proportion of the market for cars of this size in the United States, but all will be concerned at least to maintain their current positions. To what extent American production will popularize this size of car, and create a wider demand, remains to be seen. Such a trend might even help exporters to make a modest expansion of their sales there.

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The aim has been at obtaining continuous inlet and delivery processes at constant velocity, coupled with efficient compression. In order that the machine can be produced at a low cost and be thoroughly reliable, it was felt that it was necessary to break away from the normal two-rotor type of machine, which requires accurate phasing and, consequently, an expensive gear train manufactured to close tolerances.

Fig. 4. Above: The machine partly assembled showing the blades, spacing pieces and the interior of the cast aluminium alloy rotor

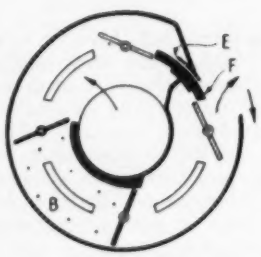


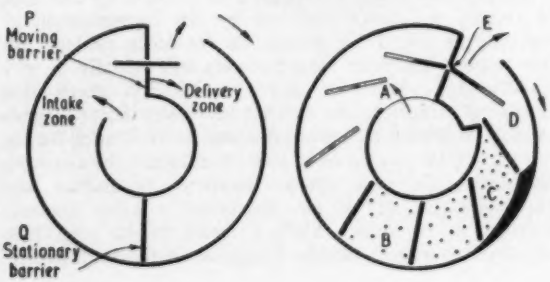
Fig. 3. Left: Later arrangement, in which the alternate blades of the rotor are replaced by stationary spacing pieces

Operating cycle

In this machine, the intake, compression and delivery processes take place in an annular channel formed in, and therefore revolving with, the rotor, although the stations at which the processes occur do not revolve. At all times this channel is divided in two, at P and Q, as shown in Fig. 1. The division P is a local constriction of the channel and the other, Q, a thin blade which is mounted on a spindle on a stationary back-plate, but can rotate about its own axis. Throughout most of the cycle this blade is positioned across the channel, to block it; then the constriction acts like a piston which, as it revolves, causes an increase in the volume of the channel on one side and a decrease on the other side of the blade: this gives rise to intake and delivery through ports on the walls of the channel. The blade is turned through 90 deg before the revolving constriction reaches it, so that it can pass through the constriction, which, in fact, is in the form of a narrow slot. In practice, it is necessary to have a series of blades, to act as partitions dividing the channel, as shown in Fig. 2, in order to give continuous sealing in the slot of the constriction.

Fig. 1. Diagram showing principle of operation of the BICERA blower

Fig. 2. Right: Modified layout giving internal compression within the unit



During rotation of one blade relative to the next, the space between them varies, as is also shown in Fig. 2, and this feature can be used to obtain internal compression. Fresh charge taken in at A is trapped in volume B and then, since this space is sealed and the channel converges, the air is compressed to volume C before this portion of the channel becomes part of the delivery space D.

Later, however, the design was modified, as shown in Fig. 3, to reduce production costs. In the new design, alternate blades were replaced by stationary spacing pieces F, which maintain the seal in the slot E. Also, the space B is of a size and shape such that its volume does not change rapidly as the blade angle changes, so compression takes place by back-flow. In low-pressure machines, this is not a great loss, since back-flow compression takes place against the stationary blade at the end of the enclosed space B, and thus does not impose any additional torque loading on the working mechanism. The compression process, therefore, differs from the back-flow compression that occurs in Roots type blowers, in which the approaching lobe is heavily loaded by it.

Mechanical construction

The geometrical arrangement is such that the blades are rotated about their spindle axes in the same direction as the rotor and at half its speed; and with this arrangement they always point to the centre of the slot E as they approach it. Thus, they are able to feed themselves through the slot. The minimum width of the slot is therefore determined, not by the geometry of the mechanism, but by the thickness of the blades at their hub. Naturally, this hub has to be strong enough to take the air loading applied over the full area of

the blade, while at the same time the deflection at the tip of the blade must not be greater than the clearance between it and the slot. On small machines, the blades are overhung from their supports, but on larger ones it would be possible to support both ends in bearings; this would permit the adoption of an appreciably greater ratio of blade length: diameter of hub.

As has already been stated, each blade is mounted on a spindle that rotates in the back-plate, and this plate also carries the blade spacing pieces, as shown in Fig. 4. A gear train giving 2:1 reduction in speed, Fig. 5, drives the blades and keeps them in phase with the rotor, which is driven directly by the mainshaft. This shaft transmits all the power required for the compression process, so the blades are free to rotate without doing work on the air.

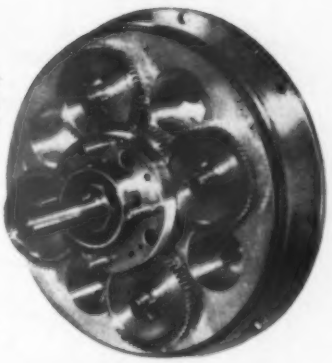


Fig. 5. View of the end of the machine, showing the whole phasing gear assembly

Design considerations

Air leakage is, of course, one of the three most important sources of loss of efficiency in rotary blowers, and machining to fine tolerances to maintain close control of clearances generally adds appreciably to production costs. In the design of the new unit, one of the objects was to simplify the profiles of components where fine clearances have to be maintained. Another aim was at making the clearances, so far as possible, independent of the accuracy of the phasing of the blades. As an outcome of this policy, the sealing faces between the rotor and casing require only simple turning, as also do the faces of the slot E and of the spacers F in Fig. 3. The slot E is of such a length that it spans the gaps between the spacers and the hubs of the blades. These hubs, of course, have a diameter equal to the width of the slot in the rotor, and since the thickness of the blades is less than this, they are free to rock about their axes as the slot passes over them, without any danger of either their tips fouling the sides of the slot or the seal being broken.

The profiles of the seals made at the ends of the blades, against the sides of the channel, are also arcs of circles,

Fig. 6. An alternative port is provided on this machine, but it is, of course, blanked off

the centres of which are on the axes of the blade spindles. Again, therefore, the blades have a certain degree of freedom to rotate about their own axes without altering the tip clearance. In the channel, the only surfaces that have to be accurately machined are those of the slot E and the walls enclosing the space B. The remainder of the channel is designed to leave adequate air space around the blades and, in practice, its shape is usually determined by the dictates of the simplicity of machining and the requirements of balancing. An approximate estimate of the swept volume of this type of machine can be obtained by multiplying the projected area of the blade, minus the cross-sectional area at its hub, by the circumference of its pitch circle. To obtain a more accurate figure it is necessary to make a small allowance for the tilt of the blades on each side of the trapped volume.

Air flow throttling losses are the second important source of restriction in the performance of rotary blowers. To a large extent these losses limit the operating speed which, in turn, determines not only the size of machine required for a given application, but also has a direct influence on its efficiency, since, by raising the speed, the throughput can be increased without any extra leakage.

With this type of blower, it has been found possible to have ports approximately equal in area to the cross section of the channel at B. Therefore, the velocity of flow throughout is comparable with that of the rotor at the mean diameter of the channel, and there is little throttling loss. In addition, the flow is assisted by taking the air in at the centre of the rotor and delivering it radially outward: this gives rise to a centrifugal effect tending to throw the air outwards in the required direction.

The third important loss to be considered is the friction of the mechanism. In the new blower the only frictional loses are those of the bearings and gears; the components effecting the compression do not need lubrication. Three ball bearings, two at one end and one at the other, carry the single rotor, and all the power required to compress the air is transmitted through its shaft. The four blades also are mounted on ball bearings and, since they run at only one half rotor speed, the frictional losses in these bearings are low, and since the gear-train loading arises only from these frictional losses, the power loss is very small. This, coupled with the fact that accuracy of phasing is not a prime requirement, permits the use of a light gear train.

The prototype blower

In Fig. 9 are shown the set of machine parts of the first prototype. At the back are, from left to right, the gear cover, back-plate, rotor, and delivery air casing. In front of these, on the left, are the four blades, and on the extreme right are their bearing housings. In the centre, is the mainshaft, with its bearing housing; and the gears are in the foreground. The rotor was of cast aluminium alloy, and was designed to be in dynamic balance. Originally, it was overhung from the mainshaft, but a better performance has since been obtained by supporting it at its outer end in a large diameter ball race mounted in the delivery air casing. At the same time, the blade design was also modified, as shown in Fig. 10. The mounting brackets for fitting the blower to an engine can be seen clearly illustrated in Fig. 6.

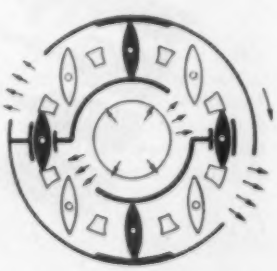


Fig. 7. A rotor arrangement giving two air cycles per revolution of the unit

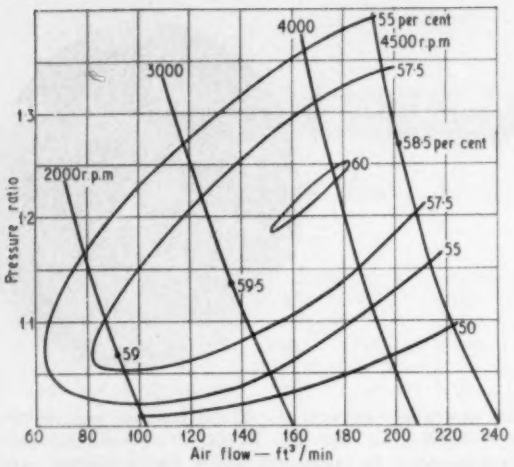


Fig. 8. Relationship between overall adiabatic efficiency, rotor speed, pressure ratio and air flow, for an outward-flow type rotor

The blades, integral with their shafts, were machined from steel strip. In the gear train there is a steel pinion and bronze idlers, and cast iron gears drive the blades. Normally the steel pinion would be hardened, but for the prototype it was not: this was to accelerate gear wear, to demonstrate that a backlash of ±1 deg, on the mainshaft, could be tolerated. The large gears dip into an oil bath, and the bearings close to the air space are of the grease-packed Neoprene-sealed type, to act as a barrier between the oil in the gear case and the air. The specification of this machine was:

Swept volume	0·0555 ft³/rev
Maximum operating speed	4,500 r.p.m.
Maximum delivery pressure	5 lb/in² (gauge)
Blade size	3×2·5 dia×0·45 in
Pitch circle diameter	5·2575 in
Casing overall size	9·25 dia×8·65 in
Weight of test machine	about 40 lb.

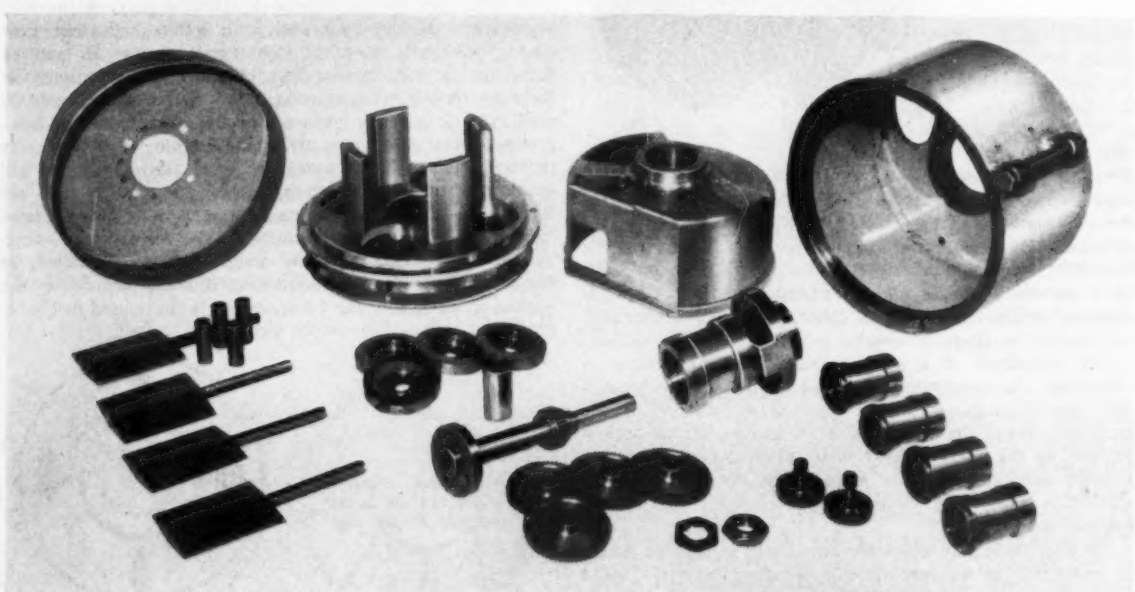
The experimental unit was designed as a relatively low pressure blower. Initially, it was intended to operate at 5 lb/in² (gauge) at 4,500 r.p.m. This gave a rotor rim velocity of 160 ft/sec and a mid-channel velocity of about 100 ft/sec. The resultant air velocity in the ports was approximately 100 ft/sec, which is lower than that commonly used in engine manifolds. For the tests, the blower was mounted on a swinging bed dynamometer, for power input measurement, and was coupled to a large meter of the air-box type, for air flow measurement.

Performance tests were carried out within the pre-set design limitations, and the results are shown in Fig. 8. The useful operating range at 55 to 60 per cent efficiency compares favourably with that of any other type of machine with comparable throughput, whether a blower or a compressor. As estimated on a basis of air temperature, the efficiencies were higher than the overall figures, because of the heat losses during compression and delivery. At low speeds and delivery pressures, these heat losses are large and the temperature differences are very small, and under these conditions the figures have little significance. However, at higher outputs, the figures obtained are worthy of note: for example, at 3 lb/in² (gauge) and above, the average temperature efficiency is seven units higher than the overall efficiency value.

Tests on an engine revealed that it would be desirable to extend the working range to 5,000 r.p.m. and 6 lb/in² (gauge) delivery pressure, to match an engine speed range of 3:1. This modification was effected by increasing the rotor rim clearances, and although it reduced the efficiencies by about three units, relative to those shown in Fig. 8, it gave a 20 per cent increase in engine torque with very little increase in minimum specific fuel consumption. These results are based on 65 hours of engine testing.

The new blower is not particularly noisy; in fact, at low speeds and pressures it is unusually quiet. Only when engine speed was increased above that which gives 4,150 r.p.m. at the blower and 4·5 lb/in² output pressure did the blower noise noticeably exceed the engine noise. A simple inlet silencer was effective in reducing the noise of the blower to a level below that of the engine throughout the operating range of the power unit, the limit of which was 2,200 r.p.m. The principal source of noise is the expansion of the small pockets of air on each side of the blade

Fig. 9. The first prototype machine, which had an inward-flow rotor, dismantled and laid out to show all its component parts



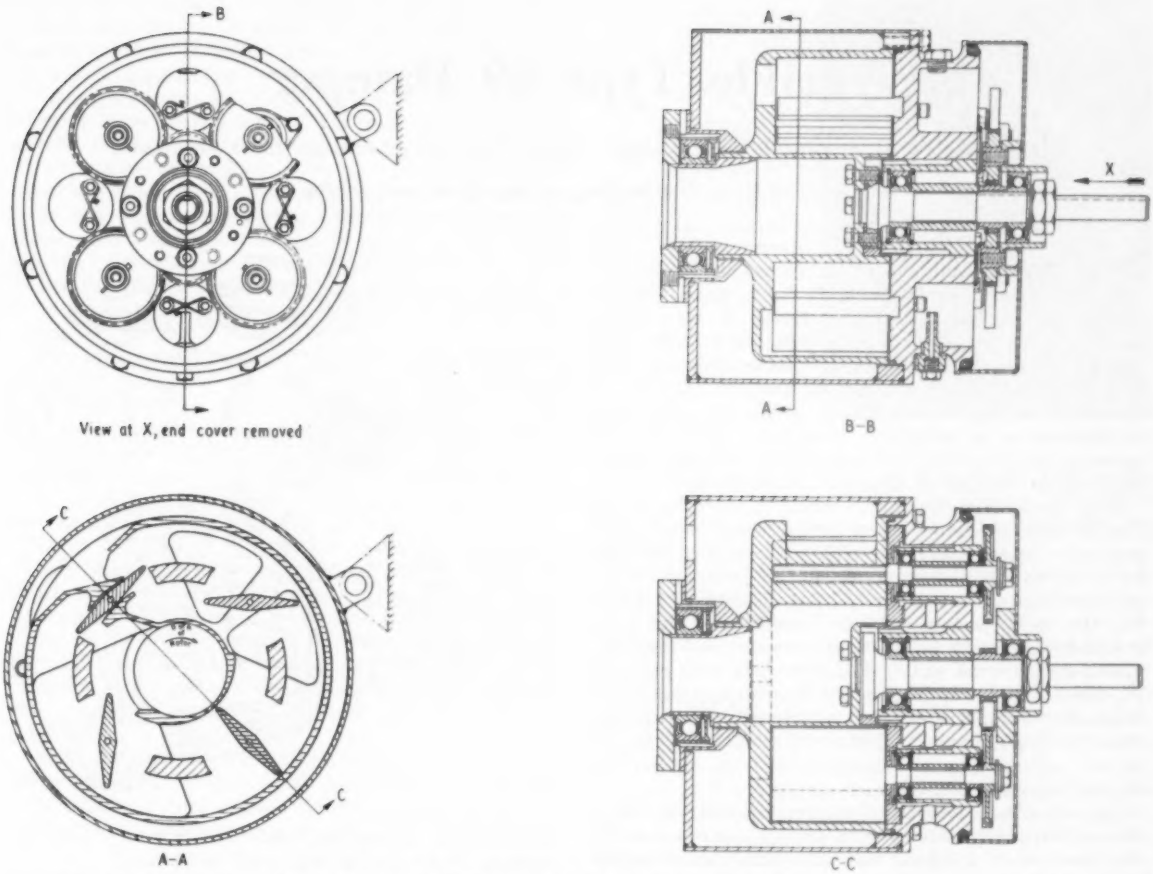


Fig. 10. In this machine, three ball bearings carry the rotor and two carry each of the blades, which have an overhung mounting

tips, as they are carried through the slot in the rotor from the delivery to inlet spaces. Rapid expansion of the air from these small pockets into the inlet space tends to excite the natural frequency of the inlet duct. By using parallel-sided blades, it would be possible to reduce noise, but at the cost of a gear train giving a high degree of phase accuracy.

Applications

Although the experimental blower is made specifically for the low-pressure charging of a four-stroke engine, it is also eminently suitable for supplying the scavenge air for two-stroke units. The outer casing forms a built-in capacity chamber, which gives the advantages of damping out pressure fluctuations and permitting a wide choice of delivery duct positions. Therefore, although this blower is larger in diameter than the bare Roots type machine, it may, in fact, be no more difficult to install because of its more favourable ducting arrangements.

The basic principle of this compression cycle can be applied to any size of machine. However, as is the case with all rotary blowers having clearances around the components that effect the compression, it is to be expected that efficiency would decrease rapidly with size for blowers whose outputs are less than 50 ft^3/min. Very small units could be made for pumping liquids, and it is thought that this principle could be applied in the design of an efficient oil or fuel pump. The cycle gives steady flow, free from fluid trapping, and is assisted by a small centrifugal head.

For the larger machines, with deliveries of perhaps more than 500 ft^3/min, it may prove to be economical to use a more complicated mechanism, in order to reduce the overall bulk per unit of swept volume, and also to take advantage of more suitable construction methods. In this way, it should be possible to make the machine simpler to produce in small numbers than is the Roots type or any of the others that require complicated rotor and casing castings. This form of design would have a rotor giving two air cycles per revolution, as shown diagrammatically in Fig. 7. An advantage of this arrangement is that the rotor would be inherently in dynamic and pressure balance. A greater number of blades would be necessary and these would be thicker relative to their lengths. Thicker blades would be easier to support on bearings at their outer ends. The bearings could be carried by a ring supported by the spacing pieces. As has already been mentioned, the use of this extra support would enable a greater blade length : diameter ratio to be adopted. In order to obtain two air cycles per revolution, the blades would rotate at the same speed as the rotor of the machine.

Anti-Freeze Fluids

TWO BRANDS of anti-freeze fluid, which are equally suitable for use in the cooling systems of diesel and petrol engines, have recently been introduced by Shell-Mex and B.P. Ltd. They are known as Shell Anti-Freeze and B.P. Anti-Frost, and they replace Snowflake Anti-Freeze, which is no longer marketed. Both are based on a formula developed by the Chemical Research Laboratory of the D.S.I.R., and they comply with the provisional British Standard Specification CY (FHC) 5100, type B. They are supplied in pint, quart, half-gallon and gallon cans, and in larger drums.

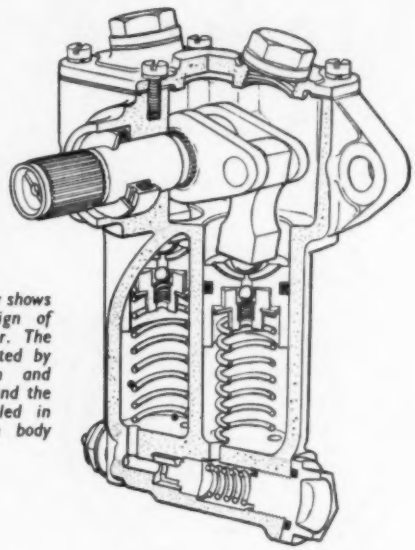
VariFlo Type 89 Damper

Description of a Simple and Robust Lever Type Replacement Unit Available with Adjustable Setting of the Rebound Valve

SHORTLY after the end of the second world war, the firm of Spax Ltd. was founded at 61 Fortess Road, London, N.W.5, to specialize in the supplying of replacement suspension dampers for cars and commercial vehicles. The magnitude of the demand for the various hydraulic units suggested that there was scope for their manufacture in competition with the established producers. For such competition to be successful, however, it was obviously necessary for the product not only to offer technical advantages but for it to sell at a strictly comparable price.

With these requirements in mind, prototypes of the VariFlo Type 89 dampers were produced and submitted to strenuous testing. The results were sufficiently encouraging for a development programme to be launched, which included intensive use of the M.I.R.A. facilities at Lindley. For this development work, an essential was the ready availability of resources for the construction and modification of experimental units. Also, it was logical to place the prototype activities in the hands of the company that would eventually be responsible for the production of the dampers. Bletchley Engineering Co. Ltd., of Weston Road, Slough, Bucks, had the requisite facilities, and the design was finalized by them for production purposes.

One of the main purposes of the development programme was to determine a range of valve settings that would enable two basic units, left-hand and right-hand, to be readily adapted to suit a large variety of vehicles. Not only would this simplify production but it would also minimize the stocks to be held by a distributor or dealer. Further research went into evolving a satisfactory means of adjusting the rebound setting, with the damper *in situ*. It was not thought necessary to make any such provision for the bump setting, because this is normally much less critical. Though they would naturally be rather more expensive than the pre-set type, the adjustable dampers would provide a means of compensating for variations in loading or road conditions. Also, they would give the small stockist a chance of meeting practically all replacement needs from a stock of only two dampers, if this was all that could be carried.



This sectional view shows the internal design of the VariFlo damper. The pistons are actuated by a rocking beam and connecting rods, and the valves are installed in the base of the body

Production of the dampers began in September 1958, and already many thousands have been sold; large numbers have gone to overseas countries. Only one basic model, the Type 89, is made at present but two others have been developed, one for heavier vehicles and the other, with a double-ended spindle, to suit the Austin front suspension. So far, no vehicle manufacturer has adopted the VariFlo as original equipment, but it is known that several have them under test. Because of the rapid build-up in the demand for replacement units, Bletchley Engineering Co. is building a new factory at Slough to ensure adequate capacity.

The VariFlo damper has a die-cast body, of Mazak 3 zinc-base alloy. It is designed to replace those makes fitted as original equipment on numerous cars and other vehicles

Construction

The VariFlo damper follows orthodox practice in having two pistons working in opposite directions in parallel cylinders of equal bore. A rocking beam arrangement attached to the operating spindle actuates the pistons, and the flow of fluid is controlled by two spring-loaded valves in the base of the body. Springs under the pistons keep these in contact with the short connecting rods between them and the rocking beam above.

To avoid any trouble from corrosion, the body is die cast in Mazak 3 zinc-base alloy, a material with which Bletchley Engineering Co. has had a great deal of experience in connection with the production of carburettor bodies. All castings are pressure tested for porosity. The upper portion of the casing houses the rocking beam, and is closed by a cover plate. It embodies two lugs for attaching the unit to the vehicle. Six screws secure the cover plate, which also is a Mazak 3 die-casting. In it are two filling orifices, symmetrically disposed on each side of the vertical centre line and inclined at equal angles thereto. The incorporation of two orifices permits a common casting to be employed

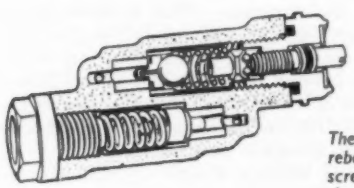
with either a left-hand or a right-hand inclined mounting.

An 80-ton steel is employed for the spindle. This is supported in two unbushed bosses in the upper portion of the body. Its projecting end has broached splines to accept the actuating lever, and there are similar splines on the length between the two bosses. The journal portions of the spindle, and the bores of the bosses, have a ground finish. To prevent the escape of oil, or the entry of dirt or moisture, a garter type oil seal is housed in a counterbore in the boss adjacent to the actuating lever. Outboard of the seal is a plain washer, which is a sliding fit on the spindle and is secured by peening the boss. Oil leakage from the inner end of the spindle is obviated by a domed disc, which is fitted into a counterbore in the body and is subsequently spread by pressure.

The rocking beam comprises two heavy-gauge steel plates of roughly diamond shape. Internal broaching is employed to form the splines in the central holes of the plates, which are pressed on to the spindle with a distance sleeve between them. At each end of the beam is a fixed spindle, pressed into holes in the plates, on which pivots one of the hardened steel connecting rods. The lower end of each rod is machined to a convex hemispherical form.

These hemispheres seat in part-spherical recesses, of slightly larger radius, machined in the tops of the pistons, which are of case-hardened steel with a ground surface. The pistons operate in pot broached bores in the body. There is no sealing ring on the bump piston, but the rebound piston carries a special, synthetic-rubber ring of X section, which prevents any parasitic bleed and minimizes the likelihood of damper fade in arduous operating conditions.

In the centre of the crown of each piston is an axial hole, the bottom edge of which forms the seating of a ball valve controlled by a light spring; the steel housing of the ball valve fits into a counterbore under the crown and is retained

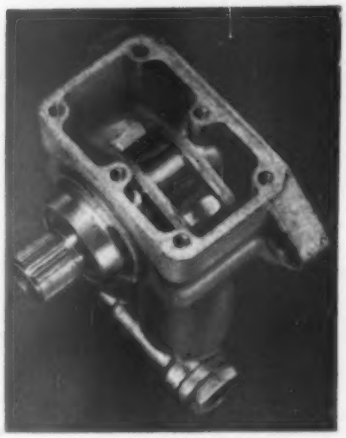


The method of adjusting the rebound valve is simple: a screw is employed to increase the loading on the spring

in position by a washer and the piston return spring. The valves are known as reassertion valves and their purpose is to keep the cylinders filled with fluid at all times. Any slight leakage of fluid past the pistons, on their downward strokes, is made good on their return strokes by the drawing of fluid through the valve from the reserve chamber, which is at atmospheric pressure. To allow easy flow of the displaced fluid between the hemispherical seating faces of the connecting rod and piston, a narrow segmental groove of relatively large radius is milled diametrically in the piston crown. The depth of the groove is slightly greater than that of the seating.

Side by side, below the bases of the two cylinders, are the bump and rebound valves. They are housed in holes drilled horizontally parallel to and one on each side of the plane containing the axes of the cylinders. The valves are connected to the lower ends of these cylinders by two pairs of vertically drilled ducts, each pair comprising the inlet and outlet from one of the two cylinders.

During the bump stroke, oil forced out of the appropriate cylinder passes vertically down through the outlet, into a horizontal duct and thence through a hardened steel metering orifice, past a spring-loaded plunger, which it



Six screws retain the cover plate on top of the damper body. Forming the rocking beam is a pair of steel plates separated by a distance sleeve

forces off its seat at the other end of the orifice, and into the valve chamber. Thence it flows up through the vertical hole to the other cylinder. Under these conditions, the rebound valve is, of course, closed.

On the rebound stroke, the valve that was opened during the bump stroke is seated by the combined forces of its return spring and the fluid pressure in the other cylinder. At the same time, the other valve is unseated, the path of the fluid through it being similar but in the opposite direction to that already described. During both strokes, the piston in the idle cylinder is returned by its spring and by the fluid transferred through the valve from the working cylinder.

So far as detail arrangement is concerned, the valves are relatively simple. A spigot extension, at the end of the hardened steel plunger, registers in the bush that forms the metering orifice. The plunger is guided by four axial ribs, cast integrally in the bore of its housing, so that the fluid can flow freely between them from one end to the other. In the pre-set dampers, the outer end of the rebound spring abuts against a screwed-in hexagon-head plug, which closes the end of the valve housing. An O-ring, for sealing, is carried in an annular groove machined in the seating face of this plug.

This arrangement has been adopted also for the bump valve of the adjustable version, but for the rebound valve, the spring and plug arrangement is rather different. The outer end of the spring of this valve abuts against a seating, which is spigoted into it. To prevent this seating from rotating, it is of hexagonal shape and is located in an axial hole, of similar section, in the end plug. A tapped axial hole in the plug accommodates an adjusting screw, which projects into the hexagon-section hole and thus regulates the depth to which the spring seating enters it. In this way, the pre-load on the control spring can be adjusted. In addition to the O-ring under the head of the end plug, there is a smaller sealing ring on the shank of the adjusting screw, and it seats in a counterbore in the outer end of the plug, to eliminate fluid leakage past the screw.

A device is incorporated to provide a positive click action for the adjustment. It comprises a pair of small balls interposed between the spring seating and the end of the screw. They are housed in hemispherical recesses in the end of the seating and they register in a diametral groove in the end of the screw every time it is rotated through 180 deg.

Yet another refinement of the adjustable unit is a simple device to avoid any tendency of the rebound valve plunger to tilt as the spring pressure is altered. It is simply a steel ball, positioned between the spring and the plunger,, and bearing in a part-spherical seating on the plunger end.

Lloyd Arabella

Interesting New German Car with 897 cm³ Horizontally Opposed Four-Cylinder Engine and Front-Wheel Drive

POST-WAR trends in small car design in Germany are exemplified by the production history of the Lloyd company during the last decade. First came a small four-seater vehicle, powered by a 300 cm³ two-stroke engine, which soon began to sell in very large numbers to a public in urgent need of personal transportation. Then, to meet growing demand for greater room, refinement and performance, the Alexander model was introduced, featuring an all-metal instead of a composite body, and a 596 cm³ air-cooled parallel-twin engine with overhead camshaft actuation of the valves. The latest model, the Arabella, carries the growing-up process a stage further and is a fully fledged small car, intended to compete with the Volkswagen and other German cars in the same category.

A satisfactory degree of engine refinement is assured by the use of a horizontally opposed four-cylinder engine with water cooling. The engine has overhead valves, operated by push rods and rockers, and drives the front wheels through a four-speed gearbox. An automatic clutch is available as an optional extra. Coil springs are employed as the suspension medium for all four wheels. The four-seater saloon body is of mainly unitary construction, but embodies a number of bolted-on components.

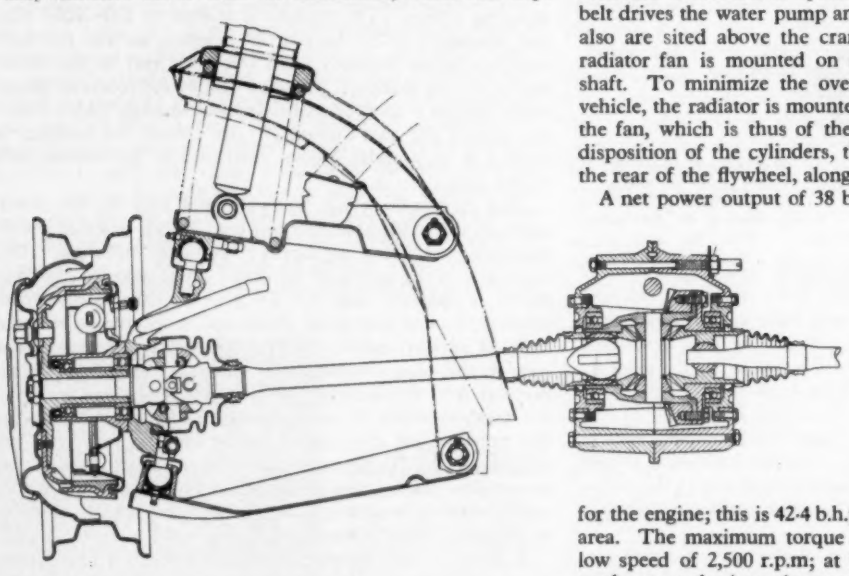
In the engine, which is mounted ahead of the front wheel axes, extensive use is made of aluminium, which has kept the weight down to the creditable figure of 175 lb. The cylinder block and crankcase unit is a two-piece aluminium casting, split vertically on a longitudinal plane through the crankshaft axis. Thus, the housings for the main bearings are split, as are the camshaft bearings, which are machined directly in the aluminium. Since there is no separate sump, attention to the cylinder bores or to the main or big-end bearings necessitates removal of the power unit for splitting.

The crankshaft is of the three bearing, four-throw type. Because of the fundamentally good balance of this engine layout, no counterweights are employed on the shaft, which is a steel forging and has a stroke of 60 mm. Since the bore is 69 mm, the swept volume is 897 cm³. The steel-backed big-end bearings are lined with lead-bronze, but the main bearings are solid aluminium, with a lead infusion of the rubbing surfaces. A gear drive is employed for the camshaft, which is mounted above the crankshaft; there are only four cams and, in conjunction with bucket type tappets, they actuate hollow light-alloy push rods.

Wet cylinder liners have been adopted. They are each located in the block by a spigot and a flange, seating on a copper ring near the lower end, and they are held in compression by the studs that secure the cylinder heads. The heads also are aluminium castings and have cast iron seating inserts for the valves, which are disposed in line and parallel with the cylinder axes. Combustion chambers of heart shape are employed and the compression ratio is 7·5 : 1. A Solex PC1 single-choke downdraught carburettor feeds through a water-heated manifold to siamesed pairs of inlet ports on the upper sides of the heads; the downward facing exhaust ports are separate.

The distributor is driven by a spiral gear on the camshaft, and is accessibly sited above the engine. Its driving shaft is extended downward to operate the oil pump. A single vee-belt drives the water pump and the generator, both of which also are sited above the crankcase. The shrouded, nylon radiator fan is mounted on the rear end of the generator shaft. To minimize the overhang of the front end of the vehicle, the radiator is mounted above the engine and behind the fan, which is thus of the pusher type. Because of the disposition of the cylinders, the starter motor is installed to the rear of the flywheel, alongside the gearbox.

A net power output of 38 b.h.p. at 4,800 r.p.m. is claimed for the engine; this is 42·4 b.h.p/litre, or 1·64 b.h.p/in² piston area. The maximum torque is 44·8 lb-ft, at the reasonably low speed of 2,500 r.p.m; at 123 lb/in², the b.m.e.p. is only moderate and, since the maximum power output figure is said to be for continuous running, it would appear that the breathing has deliberately been restricted to permit lengthy spells of full-throttle driving, without risk of overloading the engine either thermally or mechanically. At 4,800 r.p.m., the mean piston speed is only 1,890 ft/min.



The layout of the final drive and the front suspension of the Lloyd Arabella car. Constant-velocity joints are fitted to both ends of the half shafts. Ball joints are used to connect the outboard ends of the suspension wishbone links to the steering swivels

In standard form, the bell housing contains an orthodox single dry plate clutch of 7 in diameter. The Fichtel and Sachs Saxomat automatic clutch is, however, available as an optional extra and gives two-pedal control. Since the gear-

Right: Because of its good proportions and the use of 13 in wheels, the Arabella looks larger than its actual size. Its overall length is under 12½ ft and it weighs 1,529 lb

Below, right: The rear-end styling follows current trends, but the tail fins are not unduly prominent. A considerable amount of wrap-round has been given to the large rear light, and the bumper design is very simple. Although the boot is capacious, the height of skirt panel hinders access

box is to the rear of the final drive unit, the primary shaft from the clutch is unusually long. The gearbox is of the all-indirect type, with synchromesh on all forward ratios, the upper three of which have single-helical gears. Though the gear ratios are widely spaced, at 1·09:1, 1·57:1, 2·53:1 and 4·77:1, the intervals would appear to be well chosen. Reverse gear has a ratio of 4·54:1. A steering column gear shift lever is employed. To match the crankcase, the aluminium casing of the gearbox is split in the longitudinal, vertical plane, so the bearing housings are divided.

The pinion of the spiral-bevel final drive unit is formed on the forward end of the gearbox output shaft. A ratio of 3·87:1 has been adopted, and the crown wheel carries a two-pinion differential of normal design. Constant-velocity universal joints are fitted at each end of the half-shafts. These joints are designed and manufactured by Lloyd, and are shown in the accompanying illustration. The use of a spade-and-slot coupling on the inner joints avoids the necessity of sliding splines.

Double transverse wishbone links and coil springs are used for the front suspension, and ball joints are employed to connect the ends of the wishbones to the steering swivel members. The swivel axes, which have considerable lateral inclination, pass through the geometric centres of the outboard universal joints. As on the B.M.C. ADO 15 cars, described in last month's issue of *Automobile Engineer*, the springs and dampers are positioned above the upper wishbones, to avoid interference with the drive shafts. The upper ends of the springs abut against upswept extensions of the fabricated brackets that carry the suspension link pivots. Welded on top of these extensions are tubes that contain the upper ends of the telescopic dampers and carry their top anchorage bolts.

The steering mechanism is of the rack-and-pinion type. Since the rack is mounted behind the engine, the steering arms trail from the swivel members. An interesting feature of the design is the provision made to avoid the application of bending loads to the rack: the track rods are not connected directly to the rack, but to an auxiliary transverse member linked to it. The turning circle is 30 ft and movement of the steering from lock to lock requires 3¼ turns of the wheel—this is relatively low gearing for a small car, but presumably is necessary on account of the front-wheel drive layout.

Each rear wheel is carried on a trailing wishbone having widely spaced pivot bearings. The outer bearing of each wishbone is mounted slightly ahead of the inboard bearing, the consequent angle of the pivot axis providing a degree of roll over-steer to counteract the inherent under-steer characteristics of the front drive layout. To give the necessary rigidity, each wishbone has box-section arms, braced near their apex by a channel-section member, which forms

the lower abutment for the coil spring. The telescopic dampers are mounted behind the axis of the wheels and their upper ends have sandwich-type mountings on the wheel arches. A straight anti-roll bar links the outboard pivot bosses of the wishbones.

The hydraulic brakes operate in linered aluminium drums having radial cooling fins and a track diameter of 7·9 in. A lining width of approximately 1¼ in is specified; the lining area of 102 in², or almost 150 in²/ton kerb weight, is ample for a car of this type. The 13 in diameter wheels have 3·5J rims and are fitted with tyres of 5·20 in section.

An unusual underbody construction has been adopted, in which a platform is integrated with a tubular backbone member, which is of 4 in diameter and has a wall thickness of 0·093 in. At its front end, the tube is welded to a forked box-section member, the arms of which extend forward on each side of the engine sump. The forked member is bridged at the extreme front and carries the attachment brackets for the various front suspension components. A transverse, arcuate tube sweeps over the engine to brace these brackets and to form the mounting for the radiator. Each arm of the forked member is connected by an oblique, horizontal tube to the adjacent body sill.

Extensive swaging, or fluting, of the platform portion of the structure enhances its stiffness, and there are three box-section cross members between the scuttle and the rear seat pan. Integration of the scuttle and wheel arches with the platform makes an important contribution to the rigidity of the structure. The inboard pivots of the rear suspension are carried on the end of the backbone member, while the outboard pivots are mounted on longitudinal box-section members beneath the floor. These members are extended

rearward to support the boot. Although this underbody structure is undoubtedly extremely stiff, it is relatively heavy and embodies a large number of welded joints.

Whereas the main portion of the body, which is made only in two-door form, is integral with the underbody, the roof, rear quarters, wings and scuttle panelling are bolted in position, for ease of replacement in the event of damage. Although the body has pleasant lines, in modern style, it has not any marked individual characteristics. The roof pressing is rather more domed than on many contemporary cars and, though the rear light is of the wrap-round type, the windscreen has only a moderate single-plane curvature, to provide easy entry to the front seats. A curved line is pressed in the side panels, and its stepped form tends to be out of harmony with the main lines of the body. The tail fins rise slightly above the natural waistline of the car, and have an appreciable rearward rake to their trailing edges.

Since the spare wheel and the 7¾ gallon fuel tank are carried under the floor, the boot is unobstructed and of satisfactorily large capacity. However, there is a relatively high skirt panel over which the luggage has to be lifted when it is loaded or unloaded. This skirt is necessary because of the absence of a stiffening bulkhead behind the rear seat, the squab of which hinges forward as a secondary means of access to the boot, or to accommodate unusually long articles.

The doors open through 90 deg, and provide an entry width of 34 in. For the benefit of rear seat passengers, the squabs of the individual front seats have an oblique hinge axis, so that they swing inward when tilted forward. Upholstery and trim are in a mixture of plastics coated and woven plastics fabrics. The facia panel is a plastics moulding and incorporates an open parcels shelf and a glove box. A small cover in the panel can be removed to give access to the electrical fuses. Heating and demisting equipment is included in the standard specification.

According to the manufacturers, the major dimensions of the Arabella are as follows: wheelbase, 7 ft 2·6 in; track, 3 ft 11¼ in; ground clearance, 7 in; overall length, 12 ft 5·6 in; overall width, 4 ft 11 in; overall height, 4 ft 6·6 in. The kerb weight is 1,529 lb, with 55 per cent on the front wheels.

Testing Ball Bearings

AN ELECTRONIC instrument that was originally developed in America, by the Barden Corporation, is available in Great Britain. It is the Smoothrator, which not only tests the surface finish of raceways and balls, but also detects major causes of bearing malfunction, such as dirt, other contamination or damage. Although it has not been designed as a torque tester, it is claimed nevertheless to produce test results which indicate the usefulness of bearings for exacting gyro gimbal, synchro and other low torque applications. For applications requiring low noise or vibration levels, Smoothrator test results are said to be objective, reliable guides to bearing performances in completed assemblies.

Operation of the instrument is extremely quick and simple. No interpretation of the tests is required from the operator; results are read directly in units related to limits established by the bearing user and are not effected by differences in the operator's technique, skill or judgment. A test time is 30 to 50 seconds, giving an average yield of 60 to 100 bearings per hour. The unit is easily carried and can be set up in 10-15 minutes, including calibration. Its only requirements are a power source of 230-250 volts, 50 cycles—a voltage regulator may be needed if the voltage fluctuates beyond this range—and fluorescent lighting to provide the stroboscopic effect that indicates bearing speed.

Testing ball bearings with the Smoothrator, a simple, electronic device

Functionally, the advantages of employing this new unit are as follows. Reduction in assembly costs—preassembly testing of bearings screens out those likely to cause rejection of completed assemblies; identification of faulty bearings—damaged bearings in rejected assemblies are separated from those which may be cleaned and re-used; location of trouble sources—tests on bearings at various assembly stages can be used to pinpoint sources of dirt, contamination or damage which may have occurred during assembly or dismantling.

Propeller Shaft Lubrication

UNTIL recently, the recommended lubricant for Hardy Spicer propeller shafts and universal joints was oil, SAE 140 for normal use and SAE 250 for tropical or particularly arduous conditions. However, some time ago it seemed to the manufacturers that some of the improved greases that have been developed might be entirely suitable for these duties. In consequence, a long-term research programme was undertaken and led, about a year ago, to the decision that the heavier shafts in the range, the 1600 series and upwards, should be lubricated with an 8 to 12 per cent lithium-base grease containing a minimum of 3 per cent extreme-pressure additive.

Since greases, of this type, with the E.P. additive are less generally available than those without, further tests were carried out, using non-E.P. greases, with satisfactory results. In consequence, it is now recommended by Hardy Spicer that grease be used to lubricate all shafts and joints made by them. The grease employed must meet the following specification: base, lithium; soap content, 8 to 12 per cent; penetration, worked, 265 to 295; N.L.G.I. classification, 2. Given in the accompanying table is a list of the proprietary greases tested and approved by the company.

Manufacturer	Type	Manufacturer	Type
Esso Petroleum Co. Ltd.	*Esso Multi-purpose* *Esso T.S.D.1110*	Petrofina (G.B.) Ltd.	*Fina Lithium E.P. Grease*
Gulf Oil (G.B.) Ltd.	*E.P. Grease*	Regent Oil Co. Ltd.	*Marfak 0* *Marfak 2HD*
Edward Joy and Sons Ltd.	*Super Lithium Filtrate* *Super Lithium Filtrate E.P.3*	Shell-Mex and B.P. Ltd.	*Shell Retinex A* *Shell Alvania 3* *Shell Alvania E.P.2*
Mobil Oil Co. Ltd.	*Mobilgrease Larital No. 2* *Mobilgrease M.P.* *Mobilux No. 2*	Sternol Ltd.	*Ambroline L.H.T.* *Sternoline R.N.9*
Oiline Refining Co. Ltd.	*Oilith 3*	Valvoline Oil Co.	*F.V.73 E.P.*
		Edgar Vaughan and Co. Ltd.	*Evco H.S.N.*
		C. C. Wakefield and Co. Ltd.	*Spherol H.P 2* *Castrolease L.M.*

Recent Foreign Publications

Brief Reviews of Current Technical Books

Lager und Schmiertechnik (Bearings and Lubrication)

In German, by Reinhold Kamps, Dipl. Ing. Dr.

Düsseldorf: VDI-Verlag G.m.b.H. 1957. 7×9¼. 380 pp. Price D.M. 38.

The subject of bearing design and performance has become increasingly important in recent years. This is to some extent owing to the rapid development of the high-speed gas turbine and the ever increasing loads and speeds of reciprocating engine components. Much valuable research data has been published by Bowden and his co-workers in this country and by Vogelpohl in Germany, and a great deal of work has also been carried out in the U.S.A. and the U.S.S.R. Apart from this, a number of books on roller bearings have been published by manufacturers, so the first impulse is to exclaim: "What, another book on bearing problems!" However, on glancing through the book, this reaction is soon replaced by a much more favourable one. The author is an experimental and development engineer with one of the largest and best-known Continental vehicle manufacturing organizations, and the book is written against the background of his experience. It therefore will be of particular interest to design and research workers in this field.

After a brief introduction, the author deals with sliding friction and the numerous variables affecting its magnitude, the pressure distribution, heat flow characteristics, the effects of materials, oils, finish and temperatures. This section concludes with considerations relating to the composition and behaviour of lubricating oils. The second major section is devoted to sliding bearings, with particular reference to lubrication problems, the resultant design solutions and how they affect requirements in respect of the lubrication systems and their components, such as pumps, filters, the oils themselves. Also, the bearing materials, clearances, bearing wear and heat transfer characteristics are considered. Next, representative bearing designs are discussed with particular reference to the effect of component elasticity on pressure and temperature distribution. To the designer, one of the most useful sections of the book is that devoted to the calculation of bearing sizes. This covers some 25 pages and deals in considerable detail with the determination of bearing sizes, with particular reference to heat transfer problems. This first part of the book occupies about 180 pages.

Then follows an equally important part, of approximately 100 pages, devoted to rolling element bearings, written mainly by Dr. Ing. H. Perret. Here the mechanics of rolling friction, and the important subject of stress distribution at the point of contact and its effect on the bearing capacity are considered, together with the incidence of load changes and the distribution of the load at the bearing circumference. The important subject of bearing life expectancy is dealt with on the basis of the latest knowledge, and endurance test results and the methods of life expectancy determination are explained at some length. After considering the microgeometry of ball bearings, the author deals with the kinematics of rolling as well as with its elasto-kinematics, including the important vibrational aspects, the intermittently varying rolling velocities and the position of the roller path in the races. Production methods, and materials used for the manufacture of roller bearings are briefly considered next, and this section is followed by a general description of bearings and some examples of their use. Following this, is a section devoted to the various modes of bearing loading, the resultant choice of bearing seating, bearing clearances and their determination, and the possibility of axial deformation of bearing races. This section is of particular interest to vehicle and engine designers, whilst the section on high-speed bearings is of interest to designers and users of turbochargers and gas turbines. Problems of lubrication, the appreciation and recognition of typical defects and bearing wear are also considered somewhat briefly. In this section, a more detailed treatment would be of greater assistance to development engineers. The method of determining bearing sizes, on lines developed by the author, is of considerable interest. This method is illustrated with a number of worked out examples of practical problems.

The third section of the book is devoted to design considerations relating to engine bearings, the loads imposed on them, their design as exemplified by up-to-date layouts an a summary of sizes culled from representative engines. Finally, a section of the book deals with bearings for high-speed applications such as superchargers and gas turbines. The whole work is profusely illustrated. There are some 130 references, and these should facilitate further and, if necessary, more detailed study of various aspects of bearing design and performance, which the authors could only touch upon owing to the necessarily limited space available.

Kolbenringe (Piston Rings) Vol. II, Operating Performance and Testing.

In German, by Carl English, Dr. Ing.

Vienna: Springer-Verlag. 1958. 7×9½. 331 pp. Price D.M. 62.

The profusely illustrated second volume of this monumental work, which no doubt will become the standard source of reference on this important subject, is of considerable interest to engine designers and operators. The action and features of the piston ring in acting as an effective gas seal are considered in detail, and current design trends, in terms of ring assembly arrangements, are dealt with in a masterly fashion with the aid of useful tabulated data. The choice of ring materials and the effects of installation conditions are also considered in detail, and again the clear and lucid exposition will be much appreciated.

Since temperature effects play a great part in the life of piston rings, the influence of heat stresses and the resultant deformations and loss of elasticity are fully dealt with in a chapter occupying some 26 pages, and this is followed by 100 pages devoted to piston ring wear. In respect of wear, the author considers the effect of chemical attack, electrolysis, surface fatigue, etc., and then concentrates on the wear resisting characteristics of different ring materials, the matching of ring and bore materials to secure best results, the influence of ring surface finish and lubrication, the shape of worn ring profiles, and the effects of ring height, ring temperature and engine loads on wear. He also discusses wear due to starting and to air contamination, and the properties of chromium plated rings, and the effect of fuels and lubricants are also covered.

The interrelation between piston ring performance and oil consumption claims the next 30 pages, in which the author considers the effects of factors such as viscosity, engine speed and load, the action of scraper rings and oil pumping by ring action. The important subject of ring sticking is considered in the following chapter, which occupies 25 pages. In this chapter, the wealth of information provided by the author is both welcome and impressive. The causes of ring fractures are considered as a fatigue problem and, in addition, some space is devoted to the contribution of such factors as temperature. lubricants, ignition, fuel and materials. A chapter on ring replacement concludes the first part of the book.

The second part, 82 pages, is devoted to ring testing and relevant investigations. It covers ring tension and pressures. modulus of elasticity, tensile and bending strength and hardness, dimensional checking. pressure distribution and hysteresis curves. This part is profusely illustrated with reproductions of photographs and drawings of rings; the results of tests are also given. An appendix includes reproductions of some 115 metallographic etchings of piston ring materials, and the volume concludes with a bibliography of 140 references.

The thoroughness, clarity and patience with which the author has dealt with this important subject, the enormous amount of valuable information presented in a logical, unambiguous and most helpful manner, form a valuable and comprehensive text and reference book, and will place designers, manufacturers and operators greatly in the author's debt. This is indeed a masterpiece, the importance and usefulness of which will be appreciated for a long time to come by all those who design, build and run internal combustion engines, and by component manufacturers.

Untersuchungen zur Erhöhung der Sicherheit den Kraftfahrzeuge durch Verbesserung der Stabilität und der Fahrtrichtungshaltung. (Investigations to Increase Vehicle Safety by Improving Stability and Directional Control.) Deutsche Kraftfahrtforschung und Strassenverkehrstechnik, No. 122.

In German, by H. Gebelein, Dr. Phil., H. Hahn, Ing., and G. Schlick, Dipl. Ing.

Düsseldorf: VDI-Verlag G.m.b.H. 1959. 11¾ × 8¼. 29 pp. Price D.M. 19.30.

The problems relating to vehicle stability, with particular reference to wind action, have for some considerable time formed the subject of extensive investigations, notably at the Research Institute for Automobile Engineering at the Technical High School at Stuttgart, at first under Prof. W. Kamm and later under Prof. P. Riekert. This most recent report is a summary of two earlier ones prepared by the Battelle Institute at Frankfurt at the request of the Federal Ministry of Transport. In a very lucid and clear introduction, G. Leunig deals with the fundamental aspects of vehicle stability, with particular reference to the effects of wind pressure and how it affects the vehicle shape required to ensure stability. This is particularly interesting in view of the trend towards ever increasing speeds of road vehicles.

The report is divided into three parts. In the first, fundamental considerations relating to lateral forces and dynamic stability are dealt with, the effect of the main variables being illustrated by means of three worked out examples, with reference to a previous publication "Directional Stability of Motor Vehicles", by Kamm, published by the Stevens Inst. of Technology at Hoboken. Attention is drawn to the beneficial influence of suitably proportioned fins used to improve lateral stability in the face of wind gusts. The decay of vehicle deviation due to extraneous forces is considered on the basis of distance, and detailed consideration is also extended to the lateral acceleration of vehicles due to the action of side winds.

In the second part of the report, the results of wind tunnel tests carried out on car and omnibus models at Stuttgart are discussed. Here, too, particular attention was paid to the possible benefits that might accrue, so far as vehicle stability is concerned, by the use of suitably shaped and positioned fins at the rear of the roof. The results are presented in some detail and should be of considerable value, particularly in connection with operation on fast motor roads.

The last part of the report deals with the mechanical aspects of the fundamentals of vehicle stability. Among the subjects dealt with are the lateral guiding forces, air forces, equations relating to vehicle motion, the evaluation of the coefficients of these equations, and general considerations derived from the fundamental equations of motion, with particular reference to the dynamic stability of road vehicles. Attention is also drawn to the importance of shock-like disturbances caused not only by sudden wind gusts and by intermittent screening afforded by trees, buildings, bridges, cuttings, etc., but particularly when meeting other vehicles or when overtaking.

This is an important and welcome report dealing with a subject that has not always received the attention due to it. The recent development of fast motor-ways has stressed the importance that must be attached to vehicle stability. It is in this respect that the report will be found of considerable assistance to designers whose interest is to ensure improved safety, so far as the operation of cars and buses at high speeds is concerned, with a view to contributing a share to the overall effectiveness of motor-ways.

Untersuchungen über dynamische Bodenkräfte schwerer Kraftfahrzeuge (Investigations of Dynamic Road Loads due to Heavy Vehicles). Deutsche Kraftfahrtforschung und Strassenverkehrstechnik No. 120.

In German, by Otto Bode, Prof. Dr. Ing. and Paul Ochner, Dr. Ing.

Düsseldorf: VDI-Verlag G.m.b.H. 1958. 11¾ × 8¼. 30 pp. Price D.M. 16.90.

This report is the one hundred and twentieth of a series on work mainly concerning road vehicles and their components. The venture started in 1938 and terminated in 1944 with report No. 83, and was resurrected in 1955, since when the reports have appeared at a steady rate of about twelve per year. Of those published so far, about one half are from Prof. Bode's Institute.

The work, carried out mainly by the research institutes of various Technical High Schools, appears recently to be concentrated on vehicle ride dynamics and braking, although attention is also given to engine and noise problems and even to the action of magnetic track-brakes used by tramways. By and large, the Federal Ministry of Transport is the sponsoring authority, although on occasions the Ministries of the *Länder* also act as enlightened sponsors of research for the benefit of the taxpayers. Vehicle and accessory makers, as well as transport operators, provide either direct or financial aid, or both.

All the work dealt with in the report under review was carried out by the staff of the Institute for Road Transport at the Technical High School at Hanover, at the request of the Ministry of Economics and Transport for Lower Saxony, whilst financial aid was provided by the Federal Ministry of Transport. One vehicle firm provided a heavy lorry and a driver, as well as some financial aid, two firms supplied trailers, one provided experimental tyres, another made some experimental gear, free of charge, whilst further financial aid was made available by the Rubber Manufacturers Association and by one other vehicle manufacturer. These facts are mentioned merely as an indication of the pattern of co-operative research effort sponsored at Universities jointly by Ministries and industry.

Static and dynamic loads imposed by the wheels on the road depend upon vehicle design parameters such as spring stiffness, mass distribution, damper and tyre characteristics as well as upon vehicle speed and road characteristics. The fundamental interactions can be considered on the basis of certain simplifying and idealizing assumptions along the lines indicated by Lehr, Marquard and others, but since the actual pattern encountered in service is of a random character, experimental investigations are unavoidable, particularly if a clearer appreciation of the whole gamut of forcing oscillations and the resultant vehicle response modes are to be the ultimate goal.

The purpose of the investigations dealt with in this report was to ascertain the effect of variables such as speed, road condition, tyre pressure, wheel load, tyre size, springs, axle mass and damper characteristics upon the dynamic, or total, wheel load. In addition, the effects of these variables on the tangential and lateral loads were to be determined. The authors outline the equipment used, the method of application and the evaluation of the test results carried out with a fully laden 15 ton G.V.W. lorry and a fully laden 16 ton G.V.W. trailer,

Next, the test procedure is described with particular reference to the variables already mentioned, and the test results evaluated and correlated by means of diagrams, or bar charts, enabling the effects of the main variables to be readily appreciated. The method of measuring the lateral forces, and some of the results obtained, are described, and the road forces are considered also on the basis of time.

In the conclusions, the effects of various design parameters on the dynamic wheel load are taken into account, and whilst some of these are along previously, almost intuitively, considered lines, others are at variance with some of the earlier assumptions. Presumably this report is a forerunner of further extensive research on this subject, with particular reference to the dynamics of the one and two degrees of freedom system, along the lines of Matsudaira and Mitschke, and the statistical considerations applied to similar problems in the railway vehicle field by Verigo and Krepkogorskv, who have done a lot of work in this connection.

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PLASTICS FOAM

Growing Importance of a Chemically Inert Substance the Potentialities of which are Probably Not Yet Fully Exploited

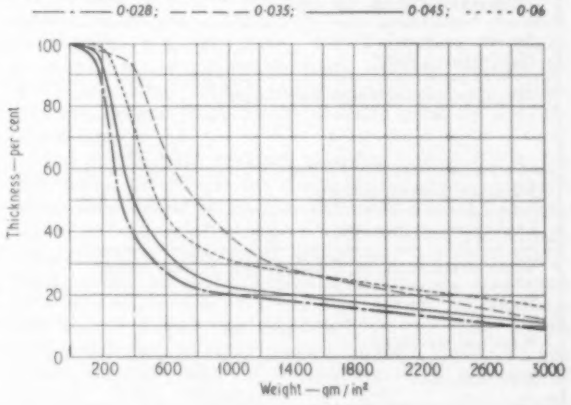
PLASTICS foam has entered the vehicle manufacturing industry in a decisive manner during the past few years; it is used in upholstery, in packaging, for shock absorption, and for heat, sound and electrical insulation. Almost certainly these uses will extend as the properties of plastics foam become more widely known. Experience of this material in the United Kingdom is relatively short, totalling only about six years. The product has a longer history in Germany, where it was first produced during the 1939-45 war; this, of course, might be expected of a country with limited access to natural rubber, for which the foam is, in several contexts, an excellent substitute.

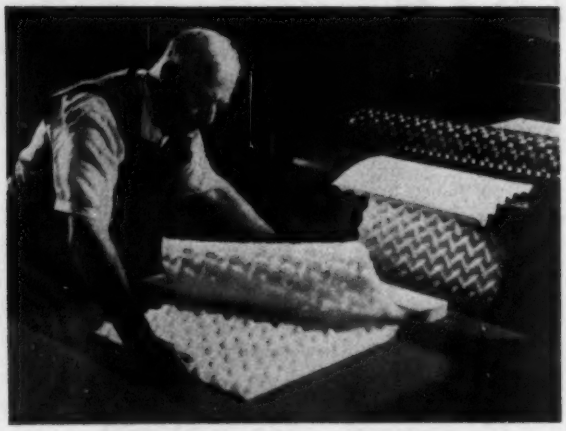
Foam material, as made by Kay Brothers Plastics Ltd. under the trade name Foamoprene, was originally a polyester foam, but is now being supplemented by a polyether grade. This latter type has one very great advantage over the polyester variety: its recovery from distortion is immediate, a property much to be desired, particularly in upholstery. Both products are manufactured by a similar process, a mixture of chemicals producing a foaming agent. Such a mix, for instance, might be sebacic acid, phthalic anhydride, succinic acid and glycerol, forming a polyester which will react with toluene diisocyanate to give off carbon dioxide as the foaming agent; this particular combination will produce a rigid polyurethane foam. For flexible foams a different polyester is used, and foam density can be reduced by adding water, which reacts with isocyanate to provide additional carbon dioxide.

The machine producing the foam is capable of an output of 2 tons per hour; it meters the chemicals, with their wetting agents, emulsifiers, catalysts and dyes, through high pressure nozzles into a mixing head, where a homogeneous mixture is achieved with the aid of a mechanical stirrer revolving at 3,000 to 4,000 r.p.m. From the mixing head the resulting liquid, which has roughly the consistency of custard, is deposited in a layer up to half an inch thick on the paper that carpets the 36 in receiving tray; the mixing head moves from side to side as the paper travels beneath it. As the chemical reaction begins, the carbon dioxide is generated and causes the liquid to foam and rise, and thereafter to set to a thickness of about a foot. The highly toxic vapour is drawn off by a ventilating plant and the operators are, of course, masked. The foam can be

In the illustration at the head of this page, Foamoprene can be seen emerging from the mixing and foaming machine. It has risen and set, and is being cut to convenient lengths for storage before processing

Below: Compression of different grades of Foamoprene under load





Profiling of the sheets of foam is effected by bisecting them while they are distorted between interlocking rollers the patterns on which are built up by the assembly of light-alloy toothed rings to form the peripheries of the rolls, which can be seen in the background in this illustration

handled after 30 minutes and can be worked after 24 hours. As it leaves the machine, it is cut to manageable lengths —usually 18 ft—by a band knife and stacked in this mattress-like form, in which it demands considerable storage space. Thus, it becomes the raw material for the secondary processing into the finished shape.

Four densities are produced as a normal range, although requirements outside this range are satisfied. The four have specific gravities of 0·028, 0·035, 0·045 and 0·055, from super-light to heavy, and the density is held to within plus or minus 0·005 gm/cm^3. With a specific gravity of 0·028, the bulk density is 1·5 lb per cubic foot; thus, plastics foam is one of the lightest materials available. The natural colour of the polyester foam is white, and that of the polyether is grey; the polyester can be dyed a number of colours, which are water-fast but which occasionally prove susceptible to browning under prolonged exposure to daylight. Dyeing of the polyethers is at present still under development.

Foamoprene is unharmed by water, air, petrol, grease, detergents and disinfectants, and it will resist weak acids and alkalis and most solvents; it will stand dry heat up to 120 deg C (248 deg F) over long periods, and it can withstand boiling or steaming for short periods. Insect pests do not spoil it, nor will it support bacterial growth. It can be flame-proofed, or made specially tear-resistant.

The tailoring of this material is largely accomplished by band knives, although it is obviously amenable to the tools and processes that are used with fabrics, such as stitching, scissoring, stapling and stamping. The thickness range is from $\frac{1}{8}$ in to 1 ft, the thin sheets being those which have been extensively used as a lining for p.v.c. in car upholstery applications over the last year or two, and which give an astonishing feeling of luxury that makes it difficult to detect the synthetic nature of the p.v.c. In the larger thicknesses the material is used for the crash padding for facias, seat and squab interiors, internal door trim, sealing and insulation of all types. Electrically, its resistance is high.

It is in upholstery that the profiling of the material becomes of value, increasing comfort and reducing weight and cost; profiling is also used for decorative purposes. The process is simple but ingenious. Interlocking rollers are made up of toothed light alloy rings, the disposition of the rings depending on the pattern required, and the degree of interlock determining the depth of the profiling. Flat blocks of foam pass through the rollers and their thickness is bisected by the band knife during the period of distortion caused by the rollers; when the foam resumes its natural shape the straight knife cut will have produced the required profile. One of the newest applications of the profiled polyether foam is in the seats of the Triumph Herald.

Although upholstery is mostly used internally, the amenability of the foam to washing and dry-cleaning is useful in this context. The material can also be disinfected, and can be autoclaved or stoved. It can even be screen printed, sprayed, electrostatically flocked or heat embossed if decorative finishes are desired, and buffing will produce almost any shape. In turn, the foam itself can be turned into flock, which means that there is virtually no waste in secondary manufacture.

Only in one or two respects does the polyether foam,

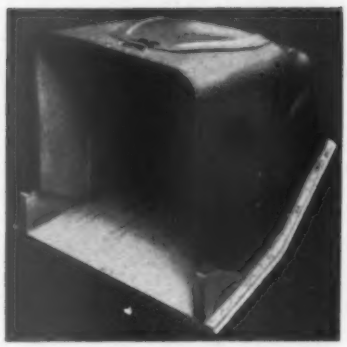
Table: PROPERTIES OF DIFFERENT GRADES OF POLYESTER FOAM

	Super-light	*Light*	*Medium*	*Heavy*
Compression set, per cent, under constant deflection for 70 hr at 22 deg C, 30 min recovery	12·5	11·5	10	8
Tensile strength, lb/in^2	20	25·33	30	24
Elongation at break, per cent	150	200	225	200
Specific gravity Bulk density, lb/ft^3	0·028 1·5	0·035 2·2	0·045 2·8	0·06 3·7
Maximum service temperature, deg C: Dry	142	140	135	130
Wet	112	110	110	100
Moisture content, per cent	1	1	1	1
Permanent set, per cent	4	3	2	2
Water absorption, as a multiple of its own weight	30	25	20	12
Di-electric constant	1·03	1·04	1·05	1·06
Electrical resistance	Better than 50 Megohms			
Pore size, mm	1	0·7	0·3	0·2

known as Foamoprene E15, differ from the polyester. The immediate recovery from compression has been quoted, and the elongation of the material at rupture is between 400 and 450 per cent; thus, Foamoprene E15 is twice as resistant to tearing as the untoughened polyester. Similarities of these materials include a compression set of 7·8 per cent and a tensile strength of 21 to 24. Properties of polyester foam are given in the table.

An adhesive process can be used to attach plastics foam to fabric or to polythene, and Kay Brothers produce a range of adhesives for sticking it to other surfaces such as wood, glass, metal or to itself. In fact, adhesives are probably the most widely used method of attachment. Foamoprene is available ready coated, with a tear-off protective lining; this lining is removed just before attachment, in the manner of that on inner tube patches. Impervious coatings, such as p.v.c. or synthetic lacquers, can be sprayed on to the surface of the foam.

This relatively new material—completely new so far as the polyether foam is concerned—has yet to expand to full utilization even in its present applications. It is directly competitive with sponge rubber and has the virtue of a very attractive appearance, owing to the sparkle of the bubble skins, which gives a crystalline sparkle to the colours.



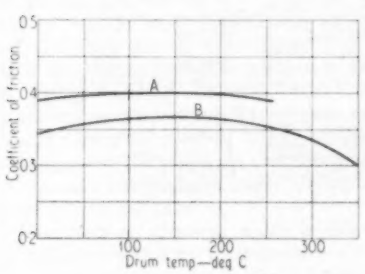
A lining of 0·028 specific gravity plastics foam, for sound insulation, inside an engine cowl for forward control vehicles

New Brake Lining

AN IMPROVED moulded brake lining material for medium and heavy commercial vehicles is now being produced by Ferodo Ltd. This new material is embodied in the AM series of linings which, in the interest of standardization for large-scale output, is applicable to a deliberately limited range of vehicles. AM linings are already being adopted, after trial, by a number of fleet operators.

This AM material has been developed in the Ferodo research laboratories. It has a high friction level combined with the unusually stable temperature characteristics shown on the accompanying graph, in which the coefficient of friction is plotted against drum temperature for rubbing speeds of up to 20 ft/sec and of 30 ft/sec. For a typical commercial vehicle having tyre and brake drum diameters of 36 in and 14 in respectively, these figures represent road speeds of about 35 and 52½ m.p.h. There is the usual slight reduction in friction level with increase in rubbing speed.

Many of the current moulded linings show an increase in the coefficient of friction with temperature up to a certain level, above which the coefficient falls rapidly as fade takes place. Such characteristics can obviously be dangerous, not only because of the rapidity of the onset of fade but also because of the virtual absence of any warning that fade is about to occur. It may be noted from the graph that, in contrast, the friction curve of the AM linings droops gently beyond its peak, and even at 350 deg C the coefficient at 30 ft/sec has not fallen below 0·3.



This graph shows the freedom from sudden fade, with increase of temperature, that is characteristic of the new Ferodo AM series of lining materials

These temperature characteristics result in a progressive but relatively small increase in pedal effort, for a given retardation, as the drum temperature becomes higher. The driver is thus given ample warning that conditions are becoming unduly severe, with the result that there is the minimum risk of high-temperature effects, such as liquefaction of hub bearing grease, failure of oil seals or damage to tyres. A further advantage of the characteristics is the promotion of greater stability under braking. Owing to the effects of side winds and manufacturing variations, the brakes on the two sides of a vehicle rarely operate at the same temperature. If the friction curve rises throughout the normal operating range of temperatures, a vicious spiral is set up: the efficiency of the brake which becomes hotter rises, so that the brake does more than its fair share of the work. Thus, the temperature difference is increased until the torque of the brake becomes sufficiently greater than that of its fellows to cause pulling to one side. A drooping curve, on the other hand, imparts automatic compensation for unbalance of temperature between the two sides.

Standard Materials

THE THIRD edition of Circular 552, entitled *Standard Materials Issued by the National Bureau of Standards*, has been issued by the National Bureau of Standards, U.S. Department of Commerce. It lists these standard materials, their certified composition and properties, also prices and purchase procedures. Included in the list are more than 550 metals, ores, ceramics, chemicals, hydrocarbons and radioactive materials. About 250 are of certified chemical composition and, of these, 100 have been prepared specifically for use in spectroscopic analysis. Among the others are those certified for properties such as acidity, viscosity, melting point, density, index of refraction and heat of combustion.

Uses of the NBS standard materials in the U.S.A. are widespread and include the checking of analytical methods or the development of new and improved methods, also the calibration and standardizing of spectrometers, *p*H meters, Geiger counters, pyrometers, viscometers and other equipment. The price of the publication is 35 cents; remittances from outside the United States must be in U.S. exchange and should include an additional 25 per cent of the publication cost, to cover postage. Orders should be sent to the Superintendent of Documents, U.S. Government Printing Office, Washington 25, D.C., United States.

Clayton Dewandre Gearbox Servo

Compressed-Air Operation Employed on a New and Compact Unit for Heavy Vehicles

To reduce the effort that has to be applied by the driver in changing gear on heavy vehicles, the Clayton Dewandre Co., of Lincoln, has introduced a servo unit actuated by compressed air from the braking system. The servo piston is mounted on a hollow operating rod, which is connected to the striker arm that actuates the selector rods in the gearbox. Within the hollow rod is the valve actuating rod, to the front end of which the gear change lever is coupled. In other words, the hollow rod and valve actuating rod assembly is interposed between the gear change control lever and the striker arm. Relative axial movement between the two rods actuates one or other of the control valves, through a scissors linkage connected to the rear ends of the rods; the valve is thus opened to admit the air to the appropriate side of the servo piston. Should the air supply fail, the gear change can still be operated manually, though with some backlash in the system.

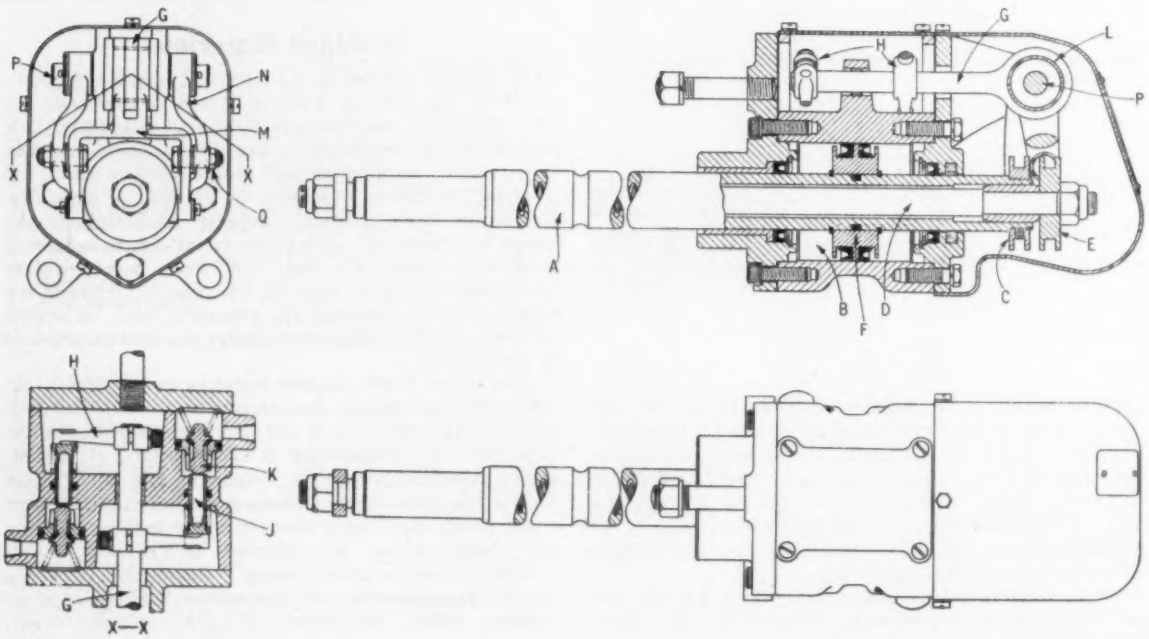
The casing of the servo unit measures only about $10 \times 6 \times 4\frac{1}{2}$ in and it forms a rearward extension of the housing for the striker assembly on top of the gearbox. Screwed on to the rear end of the hollow operating rod A, which passes through the servo chamber B, is a peripherally grooved collar C, abutting against a shoulder on the rod and locked in position by a radial grub screw. Fixed on the same end of the internal, valve-actuating rod D is a bush, which is free to slide axially in the outer member. This bush is located by a shoulder near the end of the rod, and immediately behind it is a second grooved collar E; the bush and collar are retained on the rod by a nut. A long plain bearing in the housing carries the operating rod, on which the servo piston F is located by two circlips. Round the piston are two seals, each in a peripheral groove, and air leakage between the piston and its rod is prevented by an O-ring in a groove in the rod.

Above the servo chamber is the valve rod G, which is free to move longitudinally in a bush. Two valve-operating levers H with split bosses are clamped, by means of pinch bolts, to this rod. The front lever has a downward lateral inclination to the left of the servo chamber and the other has a similar inclination to the right. These levers bear on spring-loaded tappets J actuating the air valves K, which lie one on each side of the chamber. The valves are closed by springs, and in the neutral position of the mechanism there is 0·060 in clearance between them and their tappets. When the valves are closed, both sides of the servo piston are exposed to atmospheric pressure.

On the rear end of the valve rod is formed an eye. Pressed into the eye and projecting on each side of it is a bush L, on which pivots the upper end of the forked reaction lever M. Pegs on the lower ends of the lever engage with the rear collar E already mentioned. Engaging similarly with the front collar C, on the operating rod, are two cranked support levers N. The upper ends of these levers are pivoted on a transverse fixed spindle P that has a large clearance within the bush L. Pivot bolts Q couple the reaction lever and the support levers on each side, part way up their length.

Operation

Initial movement of the gear control to the rear displaces the valve-actuating rod in the same direction, within the operating rod, so that the clearance between the two collars is increased. At the same time, the lower ends of the forked



reaction lever are moved rearwards and, since the lever pivots about the bolts Q, its upper end moves forward, taking the valve rod with it. This movement of the rod causes the right-hand valve-operating lever to actuate its tappet, and thus to operate the valve on that side, first shutting off the connection to atmosphere and then opening the valve to allow the compressed air to enter the portion of the servo chamber in front of the piston.

The air pressure moves the piston and the operating rod rearward, assisting the movement of the gear lever so long as the effort on this is maintained by the driver. When the gear has engaged, the operating rod over-rides the valve-actuating rod, which is restrained by its connection to the gear lever. The effect is to decrease the clearance between the collars and to pivot the support levers rearward about their spindle P. As a result, the pivot bolts Q also move rearward and so cause the reaction lever to swing about its lower ends, thus moving the valve rod rearward to its original position. Its movement first shuts off the supply of compressed air to the servo chamber and then re-opens the connection to atmosphere, so that the pressure on each side of the piston is equalized once more.

Forward movement of the gear lever results in a reduction of the clearance between the collars. Hence, the upper end of the reaction lever moves rearward and the left-hand valve-operating lever is brought into action. When its tappet opens the valve, compressed air is enabled to enter the servo chamber to the rear of the piston, which is thereby moved forward to follow up the actuating rod. The remainder of the operating sequence follows the same lines as those already described for rearward motion of the gear control.

The compact casing of the Clayton Dewandre compressed air servo forms an extension of the housing for the striker assembly on the gearbox

Improved K.L.G. Glow Plugs

AN entirely new type of glow plug for diesel power units has been introduced by K.L.G. Sparking Plugs Ltd. The outstanding feature of the plug is that the conventional exposed coil is replaced by a sheathed element, produced by Metropolitan-Vickers Electrical Co. Ltd. Among the numerous advantages claimed is that the outside diameter of the sheathed element is only $\frac{1}{8}$ in, compared with approximately $\frac{3}{8}$ in on the exposed coil type of plug. Thus, the new unit can be accommodated in a considerably smaller entrance hole into the pre-combustion chamber. In consequence, larger coolant passages can be embodied around the chamber, the effective volume of which is kept nearer its designed figure.

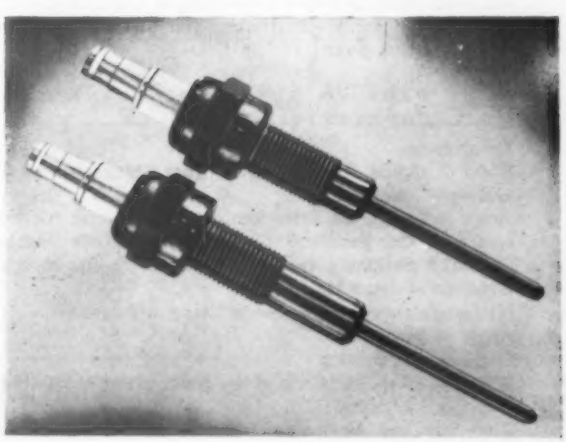
The heater wire itself, because of the shielding, is of much thinner gauge than is possible with the conventional plug. As a result, the element can be designed to work at the full battery voltage, with a reduction in current consumption, and the plugs can be wired in parallel instead of in series. With this arrangement, the engine can still be started should one plug become defective after considerable service. According to the makers, the current consumption with the sheathed element plugs is approximately half that taken by most exposed coil plugs, so that there is much less tendency for loss of incandescence when the load of the starter motor is applied to the battery.

The sheathed element, which is fully patented, has been specially developed by Metropolitan-Vickers for the purpose and embodies the experience of many years in the manufacture of this type of element, notably for domestic purposes. The sheath itself is of Inconel and the heating coil is insulated from it by magnesium oxide. To ensure good insulation and conductivity, and coaxial support of the element within the sheath, an ingenious method of construction is employed. The coil, with its connecting wire, is inserted in a tube of pure magnesium, which in turn is slid into the sheath. Then the magnesium is converted in two stages into magnesium oxide; the consequent increase in volume results in tight packing of the insulator around the coil. Good conductivity and a gas-tight seal are ensured by brazing the element into the plug body.

In spite of the insulating layer between the coil and the sheath, a warm-up time of approximately 20 seconds has been achieved, which compares favourably with that of the orthodox plug. Although heating is desirable over about 1 in of the element, to provide adequate thermal capacity, satisfactory starting can be obtained even if only the extreme tip projects into the pre-combustion chamber. The disturbance of the air flow in the chamber is therefore negligible.

At present, two types of the sheathed glow plug are available for test purposes: they have the experimental numbers E.444 and E.448, denoting long and short reach respectively. Further details may be obtained from K.L.G. Sparking Plugs Ltd., Putney Vale, London, S.W.15.

The new K.L.G. glow plugs have a sheathed element of only $\frac{1}{8}$ in diameter



BRAKE DESIGN CONSIDERATIONS

Some Notes on the Calculation of Shoe Factor

J. G. ROBINSON, M.Sc., F.R.I.C.*

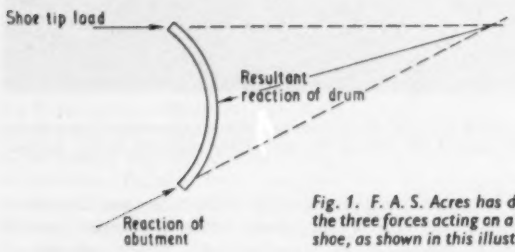


Fig. 1. F. A. S. Acres has defined the three forces acting on a brake shoe, as shown in this illustration

THIS report consists essentially of an extension of the work of F. A. S. Acres to several types of brake, including Lockheed, Girling and pivoted shoe types. Formulae are derived for calculating shoe factor, and the factors of a number of proprietary brake shoes are tabulated. The case of the Lockheed leading shoe, as used for passenger car brakes, is considered in detail, and it is concluded that the length of the lining and its position on the shoe is consistent with the requirement that the maximum shoe factor should be obtained. A brief discussion on the question of brake spragging is also included.

Key to the principal symbols used in this article

LETTERS USED FOR DIAGRAMS

A Point of application of shoe tip load
B Point of application of abutment reaction, or the pivot in the case of a pivoted shoe
O Drum centre, or centre of curvature of outer surface of brake lining
OC The line bisecting the angle AOB, or the centre-line of the shoe
OX Line joining the centre of the brake lining to point O
OY The axis perpendicular to OX
P One of the points on the line of the resultant reaction of the drum. Its position is the intersection of the construction circle with a line making an angle β with OY
Z One of the points determining the line of resultant reaction of the drum

SYMBOLS USED TO REPRESENT QUANTITIES, CONSTANTS AND ANGLES

ρ Drum radius
r Ratio of radius OA : radius of drum. Hence distance $OA = r\rho$
ϕ Angle between the lines joining drum centre to the centre of the shoe and to the centre of the lining, measured positively from the shoe centre-line in the direction of drum rotation
α Half angle subtended by brake lining at centre O
β Angle of friction
μ Coefficient of friction
l The ratio of the diameter of the construction circle: the radius of the drum
Therefore, the diameter of the construction circle $= l\rho$

l is a function of α, and for a rigid shoe, it is given by:

$$l = \frac{2 \sin \alpha}{\alpha + \frac{1}{2} \sin 2\alpha}$$

k If AB is the distance between the shoe tip and the abutment point or pivot, then

$$k = \frac{AB}{2\rho}$$

P Shoe tip load
R Resultant reaction of drum
T Tangential force acting at a distance equal to the drum radius from the centre O
F Shoe factor $= \frac{T}{P}$
ϵ Half the angle AOB
θ Angle between the direction of movement of the centre of curvature of the lining and the axis OX. θ is also used for the lower abutment angle in Appendix 3
ψ Angle that the resultant reaction of the drum makes with axis OX
σ Angle that determines the fixed point Z used in the calculation of the resultant for a pivoted shoe
δ Angle that the line of the shoe tip load makes with AB
λ Angle between the line joining point Z to the shoe tip and the line joining point Z to centre O
γ Angle between the resultant reaction of the drum and line of action of shoe tip force.

*Director and Chief Chemist, Hubert H. P. Trist & Co., Ltd.

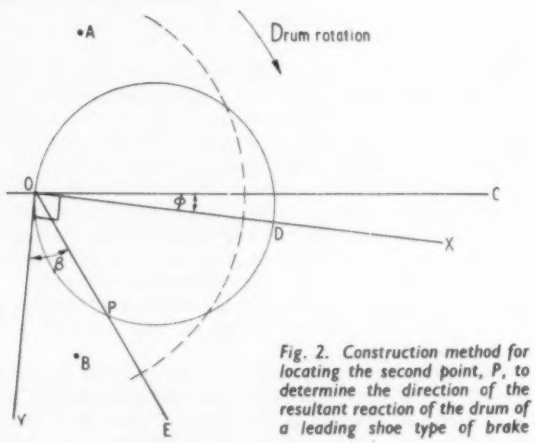
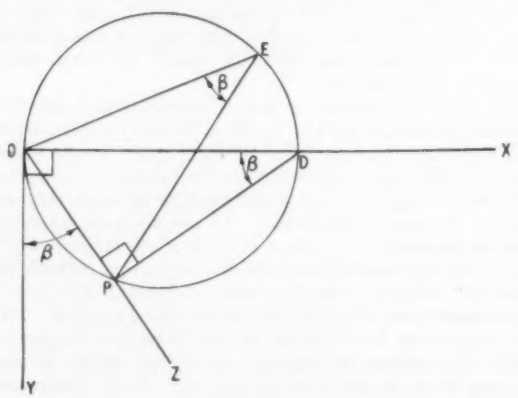


Fig. 2. Construction method for locating the second point, P, to determine the direction of the resultant reaction of the drum of a leading shoe type of brake

Fig. 3. A property of the construction circle is that any angle subtended by the chord OP is equal to β, the angle of friction



The author recently became interested in the calculation of the retardation of motor vehicles under controlled conditions. A key consideration in this type of calculation is the *brake factor* which, of course, depends on the *shoe factors*. An examination of available literature indicated that the problem had not been completely solved. The underlying theory was worked out by F. A. S. Acres[1], and has been partially applied by I. M. Waller[2]. Formulae for two types of shoe—the Lockheed parallel abutment and the pivoted types were derived by Waller. The work described in this article was undertaken to check these formulae, and to extend the calculations to other types of brake. In setting out the results, full explanations have been given, especially in the early stages.

The work of F. A. S. Acres

Acres considers theoretically the cases of the rigid shoe and the flexible shoe. He then explains the application of his results to various types of brake. In the following, an attempt will be made to give his results, but no effort will be made to consider his proofs, which are lengthy and rather difficult to follow.

According to him, a brake shoe, during application, is acted upon by three and only three forces, Fig. 1. These are:

1. Force applied at the shoe tip
2. Reaction at the abutment or pivot
3. Total, or resultant, reaction of the drum

Under these three forces the shoe is in equilibrium.

Calculation of brake factor is begun by determining the line of application of the resultant of the drum reaction. The lines of application of 1 and 2 are generally known from the construction of the brake and shoes, and the magnitude of 1 is also known. This information is sufficient for the determination of the magnitude of the drum reaction and hence the shoe factor.

The line of application of the resultant of the drum reaction is known if two points on the line can be fixed. One of these is given by the point of intersection of the lines of action of the shoe tip load and of the reaction at the abutment, Fig. 1. The position of the second point depends upon the coefficient of friction and the angle subtended by the brake lining at the drum centre. It is obtained by the following constructional method, Fig. 2.

O is the drum centre, that is, the centre of its circle

A is the shoe tip and B is the point on the abutment face through which passes the abutment reaction, or the pivot in the case of a pivoted shoe

The dotted circle represents the drum circle

OC is obtained by bisecting the angle AOB

OX is the line bisecting the arc of the brake lining. The angle XOC is a measure of the amount to which the lining

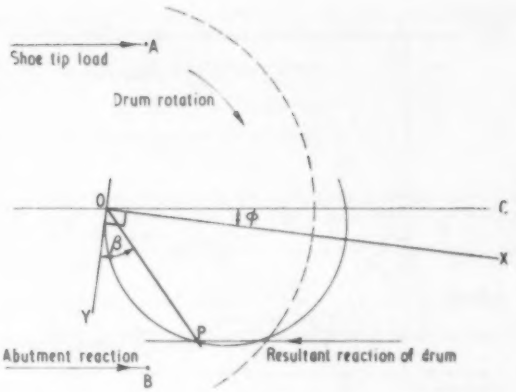


Fig. 4. With a sliding shoe of the Lockheed type leading shoe layout, having parallel abutments, the shoe tip load, abutment load and resultant reaction of the drum are all acting parallel to the line OC

is off-set on the shoe and is denoted by the symbol ϕ throughout this report

ρ is the drum radius

Let 2α be the angle subtended by the brake lining at the centre O

Also, let β be the angle of friction; then $\tan \beta$ is the coefficient of friction μ

The first step is to calculate the diameter of a construction circle, which Acres calls the basic circle.

For a rigid shoe, the diameter of the construction circle is:

$$\frac{2\rho \sin \alpha}{\alpha + \frac{1}{2} \sin 2\alpha}$$

In Fig. 2, this diameter is OD and is slightly larger than the drum radius. OD is measured along OX.

Construct a circle on OD, and draw a tangent OY to it at O. Then OY is perpendicular to OX.

From O, draw OE so that $\angle YOE$ = the angle of friction β. Then OE cuts the construction circle at P, the second point through which the resultant reaction of the drum will pass. The other point is determined by the directions of the shoe tip and abutment forces.

It is usually possible to determine the direction of the shoe tip and abutment forces by inspection. However, with a pivoted shoe, the direction of the reaction at the pivot is not known, and another approach to the problem is required. This is provided by Acres, who derived an equation for the angle that the resultant reaction of the drum makes with a given axis. The equation will be discussed when the pivoted

Table 1—DETAILS OF BRAKE SHOES FOR WHICH THE FACTOR WAS CALCULATED

Measurements	*Lockheed, Morris Minor front*	*Lockheed, Morris Oxford front*	*Girling, Ford Consul front*	*Huck, Vauxhall Wyvern leading*	*Huck, Vauxhall Wyvern trailing*
Radius of drum, ρ, in	3½	4½	4½	4 $\frac{8}{16}$	4 $\frac{8}{16}$
Distance OB, in	—	—	—	—	3 $\frac{11}{16}$
*Distance AB, in	5·29	6·75	7·12	6·75	7·18
Angle ϵ, deg	66	73	75	67	76
Angle α, deg	57½	57	55	60	57
Angle ϕ, deg	9	9	9	nil	nil
Angle θ, deg Fig. 11	90	90	60	78	—
Angle δ, deg Fig. 11	90	90	90	90	90
Angle θ Fig. 7	—	—	—	—	nil
Constant l	1·158	1·153	1·146	1·169	1·153
Constant k	0·757	0·750	0·791	0·740	0·787

**Note.—Distance AB is normally measured from shoe tip point to centre of abutment face. In the case of the Morris Oxford shoe, the initial measurement was made in this way giving k = 0·782 and this value was used for the calculations in Tables 4 and 5. Subsequently it was concluded that the shoe must pivot on the inner radius of the abutment face, giving the value k = 0·75, inserted above. The two values of k give F values varying by about 5 per cent, but the discrepancy in no way invalidates the conclusions drawn from the results of Tables 4 and 5.*

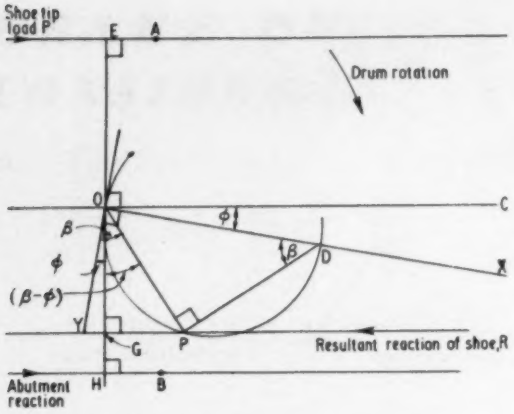


Fig. 5. Shoe factor for Lockheed sliding shoe, leading shoe layout

shoe is considered in detail. Acres also indicates, without going into details for particular types of brake, how the shoe factor can be calculated.

A property of the construction circle

In Fig. 3, OX is the axis through the centre of the brake lining; as before, OD is the diameter of the construction circle and point P is determined as already described. Join PD.

Since $\beta = \angle YOZ = 90 - \angle DOP = \angle ODP$, it is evident that $\angle ODP = \beta$.

It follows that if any point, say E, on the construction circle is selected then $\angle OEP = \beta$, since both are angles subtended by chord OP. This is an important property which will be used in the subsequent proofs in this article.

Applications

Fig. 4 shows the outline construction for a Lockheed leading shoe. Its lettering is as previously described for Fig. 2. The fixed abutment faces on the Lockheed brake are made so that the reaction on the shoe is parallel to the shoe tip force, which in turn acts parallel to the line of symmetry through the shoe, that is, to OC in Fig. 4. Since the resultant reaction of the drum balances the two other forces, it must also act parallel to OC. For a given coefficient of friction, corresponding to angle of friction β, the resultant must pass through point P. Hence the line of action of the resultant is completely determined. The derivation of an expression for the shoe factor of a Lockheed shoe is given in Appendix 1.

Acres divides brake shoes into the following types, *a* sliding shoes, *b* pivoted shoes and *c* articulated shoes. The Lockheed shoe is one of the sliding types, and has already been considered. Another, the Girling type, which has a 60 deg abutment angle, is dealt with in Appendix 3. Pivoted shoes are considered in Appendix 2.

Probably the best known type of articulated shoe is the leading shoe of the Huck brake. Acres has shown that this shoe may be treated mathematically as a sliding shoe, with an abutment reaction along the line of the link. The appropriate formula is given in Appendix 3. In Appendix 4, the case of a semi-flexible shoe is dealt with, and in Appendix 5 the case of contact at shoe ends only is discussed.

Calculation of shoe factors

The calculation of shoe factor involves measurements on the shoes and the derivation of constants l and k, as defined in the key at the beginning of this article. Measurements of a number of lined brake shoes are given in Table 1. The factors obtained over a range $\mu = 0{\cdot}3$ to $\mu = 0{\cdot}5$ are shown in Table 2. Because of differences in dimensions and the basic differences in design, the factors for different brakes were not strictly comparable. To obtain a direct comparison, a recalculation of shoe factor was made in some cases, using the same values for l and k as were found for the Girling shoe, Table 2, columns 5 and 6. The calculation for Morris Oxford front shoes is not included in the Table, since the constants l and k are similar to those for the Morris Minor.

From Table 2 it can be seen that a Lockheed type shoe, column 5, gives a slightly higher value of F than the comparable Girling shoe, column 2, up to $\mu = 0{\cdot}43$, after which the Girling shoe gives a slightly higher factor. However, the main feature of interest in Table 2 is the similarity of results, for a given μ value, irrespective of shoe design or dimensions. Table 3, shows that for the trailing shoes investigated, differences in construction have an even smaller effect on shoe factor than for the leading shoes of Table 2.

Shortened formulae for Girling and Huck shoes

The calculation of shoe factor for the Girling shoe or the primary shoe of the Huck brake is a cumbersome and tedious process, and a shorter method would be welcome. A close approximation to the shoe factor of the Girling shoe

Table 3—CALCULATION OF SHOE FACTOR FOR TRAILING SHOES

Coefficient of friction	*Lockheed, Morris Minor rear-trailing*	*Huck, Vauxhall Wyvern trailing, pivoted*	*Flexible shoe, same data as Girling*
0·30	0·459	0·460	0·464
0·35	0·502	0·511	0·510
0·37	0·518	0·530	0·527
0·39	0·532	0·548	0·542
0·40	0·539	0·557	0·549
0·41	0·546	0·566	0·557
0·43	0·559	0·583	0·572
0·45	0·571	0·600	0·586
0·50	0·597	0·639	0·619

Table 2—CALCULATION OF SHOE FACTOR FOR LEADING SHOES

Coefficient of friction	*Girling, Ford Consul front*	*Lockheed, Morris Minor front*	*Huck, Vauxhall Wyvern leading*	*Lockheed, sliding shoe, same data as Girling*	*Flexible shoe, same data as Girling*
0·30	1·028	1·168	1·132	1·111	1·121
0·35	1·329	1·496	1·388	1·409	1·429
0·37	1·459	1·641	1·529	1·536	1·569
0·39	1·614	1·795	1·667	1·672	1·719
0·40	1·702	1·877	1·737	1·742	1·797
0·41	1·781	1·962	1·819	1·819	1·881
0·43	1·956	2·136	1·965	1·969	2·059
0·45	2·142	2·323	2·111	2·127	2·254
0·50	2·699	2·835	2·592	2·555	2·822

may be made by assuming that the resultant reaction of the drum passes through the intersection of the construction circle and axis OX, point D of Fig. 12. The formula then becomes identical with that derived in Appendix 4, formula 2, provided that the value of l be calculated for a rigid shoe. This calculation for the Girling shoe, at $\mu=0{\cdot}4$, gives $F=1{\cdot}718$, as compared with 1·702 by the correct method, the difference being only about one per cent.

For the Huck brake leading shoe, the force diagram is similar to that of the Lockheed shoe, the difference being that angle θ, Fig. 11, for the Huck shoe is 78 deg, as compared with 90 deg for the Lockheed brake. Therefore, the Lockheed shoe formula can be applied as a close approximation. For $\mu=0{\cdot}4$, the Lockheed formula gives $F=1{\cdot}745$, as compared with 1·737 for the correct formula, the difference being less than one per cent.

Waller's formula for the Lockheed shoe

It should be noted that formula 5, derived in Appendix 1 for the Lockheed shoe, differs from that given by Waller, the difference being that formula 5 has *cosec* β *sec* $(\beta-\phi)$ in the denominator, where Waller has *cosec* β only. There seems no doubt that Waller's formula is incorrect. In any case, it is incomplete as it does not take angle ϕ into account. Waller's formula applied to the Morris Minor shoe, at $\mu=0{\cdot}4$, gives $F=1{\cdot}988$ compared with 1·877 in Table 2; the difference is about 6 per cent.

Effect of position of lining on shoe

On the leading and trailing shoes of most Lockheed brakes, the lining does not fill the shoe; there is a gap at the leading end but no gap at the trailing end of each shoe. Table 4 shows what would happen to the brake factor if the lining of a Morris Oxford leading shoe were displaced in stages towards the leading end, finishing up with a full gap at the trailing end. It can be seen that there is a progressive loss in brake factor with displacement towards the leading end and the loss of factor at the limit of displacement is nearly 25 per cent. It is evident that there is a good reason for the Lockheed practice of placing the lining as near as possible to the trailing ends of their brake shoes.

Effect of length of lining

Having established that one end of the lining should be fixed as near as possible to the trailing end of the shoe, the question arises as to whether a longer or a shorter lining would be more effective than the present length which, for Lockheed passenger shoes, appears to subtend an angle of about 110 deg. Table 5 shows factor against shoe length, beginning with the full length of the shoe, diminishing through the actual lining length, and going down to almost one quarter of the present length. The Table indicates that the maximum shoe factor is given by linings subtending angles in the range 100-127 deg, the factors falling off above and below this range. The present Lockheed practice appears to be 110-114 deg and is therefore midway between the limits calculated above.

Spragging with the Lockheed brake

For the Lockheed brake, the formula derived is

$$F=\frac{2l}{cosec\ \beta\ sec\ (\beta-\phi)-l/_k}$$

The condition for spragging is that the denominator becomes zero when F becomes infinitely great. This occurs when:

$$cosec\ \beta\ sec\ (\beta-\phi)=l/_k$$

For the Lockheed Morris Oxford brake, the value of l for full shoe contact is 1·153, but for the special condition described in Appendix 5, contact at ends only, the value of l becomes *sec* 57 deg = 1·836. Table 6 shows the value of k at the spragging point, for various coefficients. Spragging would be expected for all coefficients for which the k value in the Table is equal to or exceeds the actual k value of the brake, that is, approximately 0·75. For $l=1{\cdot}153$, full lining contact, the value for spragging does not reach 0·75 even at $\mu=1{\cdot}0$. Hence under conditions of full lining contact we must conclude that it is impossible to sprag the brake. For $l=1{\cdot}836$, with the ends in contact only, it will be noted that the value of k exceeds 0·75 at $\mu=0{\cdot}5$, but not at $\mu=0{\cdot}4$. Hence, for these extreme conditions, spragging is to be expected when the coefficient of the lining approaches 0·5. From these considerations, it follows that the Lockheed Morris Oxford brake should be virtually sprag-free.

Table 4—EFFECT ON SHOE FACTOR OF POSITION OF LINING ON SHOE, LOCKHEED, MORRIS OXFORD—LEADING SHOE

Total angle of shoe capable of bearing lining	132 *deg*
Angle of lining	114 *deg*
Difference	18 *deg*
Angle, between shoe centre and lining centre, ϕ =	9 *deg*

Shoe factors worked out for $l=1{\cdot}153$, $k=0{\cdot}782$

Original value of $\phi=9$ *deg. For this value, the end of the liner is on the extreme trailing end of the shoe. As* ϕ *gets less, the liner shifts towards the leading end of the shoe.*

Value of ϕ, *deg*	*Corresponding value of F for* $\mu=0{\cdot}40$	*F, as a percentage*
9·0	1·785	100·0
4·5	1·705	95·6
0	1·610	90·2
—4·5	1·505	84·3
—9·0	1·380	77·4

Table 5—VARIATION OF ANGLE OF BRAKE LINING COMMENCING WITH FULL LENGTH OF SHOE AND REDUCING IN STEPS, FROM THE LEADING EDGE—ONE EDGE REMAINS AT THE TRAILING EDGE

Angle subtended by lining, deg	*Angle* ϕ, *deg*	l	*F, for* $\mu=0{\cdot}40$	*F, as a percentage*
132	0	1·198	1·750	97·5
127	2·5	1·188	1·785	99·4
122	5·0	1·176	1·800	100·2
117	7·5	1·163	1·805	100·5
114	9·0	1·152	1·795	100·0
110	11·0	1·146	1·798	100·2
100	16·0	1·122	1·790	99·8
90	21·0	1·100	1·720	95·8
80	26·0	1·079	1·635	91·0
60	36·0	1·045	1·450	80·6
30	51·0	1·012	1·140	63·5

Note.—Value of k *used in above calculations* = 0·782.

Spragging with other types of brake

A brief consideration of the force diagrams will indicate that brakes producing resultants inclined to the shoe tip load, and this means all except the Lockheed type, will sprag at a lower coefficient than the latter type. The problem can be investigated algebraically by the method already outlined. In this connection, the shortened formulae for Girling and Huck shoes should not be used, because large discrepancies would be found at high μ values.

Note on the value of k

The value of k is found by measuring the distance between the points of application of the shoe tip load and the point of pivoting or sliding at the abutment end of a sliding shoe. It is difficult or impossible to locate the latter point with

Table 6—LOCKHEED BRAKE—CONDITIONS FOR SPRAGGING, $\phi = 9$ DEG

μ	Cosec β sec $(\beta - \phi)$	Value of k for spragging	
		$l = 1{\cdot}153$, *full lining contact*	$l = 1{\cdot}836$, *end contact only*
0·4	2·76	0·418	0·665
0·5	2·34	0·494	0·786
0·6	2·09	0·554	0·878
0·7	1·94	0·596	
0·8	1·84	0·630	
0·9	1·75	0·660	
1·0	1·74	0·665	

Note.—Spragging would be expected when k exceeds 0·75—that is not at all for full liner contact, but above $\mu = 0{\cdot}4$ for liner end contact.

any degree of precision, and in fact it probably varies from one shoe to another nominally identical shoe. The value of k is critical in the calculation of shoe factor, and actual variations in k from shoe to shoe may well explain the different wear of nominally identical shoes of a 2LS brake, which has been observed in practice.

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APPENDIX 1

SHOE FACTOR FOR LEADING AND TRAILING SHOES, PARALLEL ABUTMENT, LOCKHEED TYPE

Leading shoe

In Fig. 5, OC is the centre-line of the shoe, OX the centre-line of the brake lining, and other letters are, in general, as explained for Fig. 4, and at the beginning of this article. The shoe tip load P acting at A, together with the abutment force acting at B are balanced by the drum reaction R acting along PGY. It can be seen that the lines of action of all three forces are parallel to OC. The drum circle is not shown in the diagram, since it is unnecessary for the proof.

$$\text{Shoe factor } F = \frac{\text{Tangential force at drum radius}}{\text{Shoe tip load}}$$

$$= \frac{T}{P} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

The torque produced by tangential force T acting at radius ρ from the drum centre, is the same as that of the resultant R acting at distance OG from the drum centre.

Hence $T \times \rho = R \times OG$ ..... (2)

Considering the forces acting on the shoe, and taking moments about B, the abutment point, we have:

$$R \times HG = P \times HE \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Eliminating R from equation (2) and (3) we have:

$$F = \frac{T}{P} = \frac{HE \times OG}{HG \times \rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

However this expression for shoe factor is not yet suitable for practical application, because it contains terms that are not directly measurable, for instance HG and OG. To make the equation useful, it is necessary to introduce known quantities. The known lengths are shoe tip to abutment, AB, and diameter of construction circle, OD. As explained in the key at the beginning of this article, OD can be expressed as $l\rho$ where l is a constant and ρ the drum radius.

The length AB is conveniently expressed as $2k\rho$ where k is a constant.

It should be noted that

$$EO = OH = \tfrac{1}{2}\, HE = \tfrac{1}{2}\, AB = k\rho$$

$$\angle YOG = \phi, \quad \angle GOP = (\beta - \phi)$$

Of the lengths in equation (4),

$$HE = 2\, k\rho$$

$$OG = OP \cos(\beta - \phi) = OD \sin\beta \cos(\beta - \phi)$$
$$= l\rho \sin\beta \cos(\beta - \phi)$$

$$HG = OH - OG$$
$$= k\rho - l\rho \sin\beta \cos(\beta - \phi)$$
$$= \rho[k - l \sin\beta \cos(\beta - \phi)]$$

Substituting in equation (4)

$$F = \frac{2k\rho\, l\rho \sin\beta \cos(\beta - \phi)}{\rho[k - l \sin\beta \cos(\beta - \phi)]\rho}$$

Cancelling ρ and dividing through by $k \sin\beta \cos(\beta - \phi)$ we have:

$$F = \frac{T}{P} = \frac{2l}{\text{cosec } \beta \sec(\beta - \phi) - l/k} \quad \ldots\ldots\ldots\ldots\ldots (5)$$

Trailing shoe—Lockheed type

The diagram for a trailing shoe is given in Fig. 6, which differs from Fig. 5 in the following respects:

(1) The direction of drum rotation is reversed, the direction of the shoe tip load being unchanged.

(2) Angle ϕ is measured anticlockwise from centre-line OC, instead of clockwise as in Fig. 5. In both cases ϕ is measured in the direction of drum rotation, the lining thus being placed nearer the trailing edge of the shoe, which is the usual practice.

(3) Angle β is measured clockwise from OY, that is, its sign is reversed: this is Acres's procedure for the trailing shoe.

The working out of the expression for shoe factor is exactly analogous to that for the leading shoe, and the result is:

$$F = \frac{T}{P} = \frac{2l}{\text{cosec } \beta \sec(\beta - \phi) + l/k} \quad \ldots\ldots\ldots\ldots\ldots (6)$$

Notes

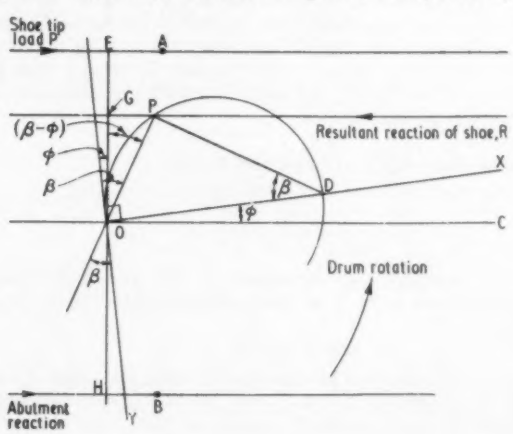
(i) Expression (6) can be derived algebraically from expression (5) by making the following substitutions:

(a) For $+T$ write $-T$ because of the reversal of the drum rotation

(b) For β write $-\beta$ because of the altered construction

(ii) The convention regarding the sign of ϕ should be noted, that the angle is measured positively in the direction of drum rotation. It is unlikely that practical cases will be found where ϕ is negative, but if they are met, the shoe factor can

Fig. 6. Shoe factor for Lockheed sliding shoe, trailing shoe layout



be found readily by substituting ϕ with a negative sign.

APPENDIX 2

The pivoted shoe

One of the assumptions made by Acres is that the application of a brake consists of "nothing more nor less than the act of displacing the centre of the shoe by a small amount in some definite direction". With sliding shoes, the direction of displacement cannot be readily predicted, but in the case of a pivoted shoe, the direction is predictable and is, in fact, at right-angles to the line joining the pivot to the shoe centre. In Fig. 7, O is the centre of curvature of the shoe, or "the centre of the shoe" referred to by Acres; B is the pivot; OC is the centre-line of the shoe; and OX is the centre-line of the brake lining. It is evident that the centre O is constrained to move round the pivot in an arc of radius BO: the amount of movement possible is, of course, very small, and the direction of movement is tangential to the arc, that is, at right-angles to the straight line joining BO.

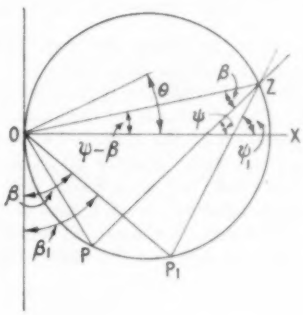


Fig. 8. Leading pivoted shoe: the resultants corresponding to the angles of friction β and β_1 are ZP and ZP_1 respectively

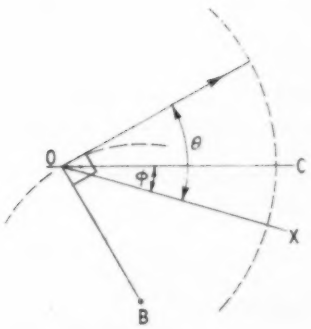


Fig. 7. Pivoted shoe: the direction of displacement of the drum centre O is indicated by the arrow

Fig. 8 illustrates the method used by Acres to find the resultant reaction of the drum on a leading pivoted shoe. All angles are referred to axis OX, which is the centre-line of the brake lining. The direction of shoe centre displacement makes an angle θ with OX. Points P and P_1 are points on resultants, corresponding to angles of friction β and β_1, the construction being as previously described. Acres shows that both, and indeed all, resultants must pass through a fixed point Z on the construction circle. The point Z is fixed by angle XOZ, which is related to angle θ in the following manner:

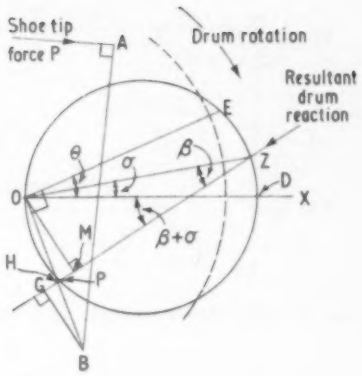
$$\tan \angle XOZ = \tan(\psi-\beta) = \text{constant} \times \tan\theta \quad \ldots\ldots\ldots(1)$$

In this expression, ψ is the angle that the resultant makes with OX. The value of ψ varies with the value of β, but $(\psi-\beta)$ remains constant. In this equation, the constant that is multiplied by $\tan\theta$ is a function of the angle of the brake lining $2a$, and for a rigid shoe:

$$\tan \angle XOZ = \frac{a-\frac{1}{2}\sin 2a}{a+\frac{1}{2}\sin 2a}\tan\theta \quad \ldots\ldots\ldots\ldots\ldots(2)$$

It is now necessary to derive an expression for shoe factor.

Fig. 9. Shoe factor for leading type pivoted shoe. The angle σ in this illustration represents the angle which in Fig. 8 was $(\psi-\beta)$. This avoids undue complication of the formula finally derived in Appendix 2



In Fig. 9, OX is the centre-line of the brake lining, B is the pivot and OE is the line of displacement of the shoe centre, $\angle BOE$ being a right-angle. Line OE makes an angle θ with axis OX.

The construction circle is drawn, with OD as diameter; point P is fixed, corresponding to an angle of friction β, by the construction of Fig. 2. Next, the angle XOZ is drawn, cutting the construction circle at Z, which is the fixed point previously discussed; the angle XOZ is derived from equation (2). Angle XOZ, which was previously called $(\psi-\beta)$, is in this case called σ to avoid undue complication in the final formula. The resultant reaction of the drum is ZP, which when produced cuts OB at H; the angle $OZP=\beta$. From O draw OM perpendicular to ZP, and from B draw BG perpendicular to ZP produced. To find the shoe factor, proceed as in Appendix 1. Using symbols previously employed, and listed at the beginning of this article:

$$T\rho = R \times OM$$
$$P \times AB = R \times BG$$

Hence
$$F = \frac{T}{P} = \frac{OM \times AB}{BG \times \rho} \quad \ldots\ldots\ldots\ldots\ldots(3)$$

In order to produce an equation for practical application, the quantities of equation (3) must be expressed in terms of known lengths and angles. The known lengths are:

$OB = r\rho$

OD = diameter of construction circle $= l\rho$

AB = length of perpendicular from pivot to line of action of shoe tip load

The known angles are β, θ and σ and various right-angles.

Equation (3) contains the ratio $OM : BG$, and from similar triangles OMP and BGP, it follows that:

$$\frac{OM}{BG} = \frac{OH}{BH}$$

It is evident that we can find OH in terms of OD if we know $\angle HOM$

Now $\angle HOM = \angle HOE - \angle MOD - \angle DOE$

$$= \frac{\pi}{2} - (\frac{\pi}{2} - \beta - \sigma) - \theta$$
$$= \beta + \sigma - \theta$$

Also $OH = OM \sec(\beta+\sigma-\theta)$
$$= OZ \sin\beta \sec(\beta+\sigma-\theta)$$
$$= OD \cos\sigma \sin\beta \sec(\beta+\sigma-\theta)$$
$$= l\rho \cos\sigma \sin\beta \sec(\beta+\sigma-\theta)$$

and $BH = BO - OH$
$$= r\rho - OH$$

Substituting:

$$F = \frac{AB\, l\rho \cos\sigma \sin\beta \sec(\beta+\sigma-\theta)}{\rho\rho(r - l\cos\sigma\sin\beta\sec(\beta+\sigma-\theta)}$$
$$= \frac{AB\, l \cos\sigma \sin\beta \sec(\beta+\sigma-\theta)}{\rho[r - l\cos\sigma\sin\beta\sec(\beta+\sigma-\theta)]} \quad \ldots\ldots\ldots\ldots(4)$$

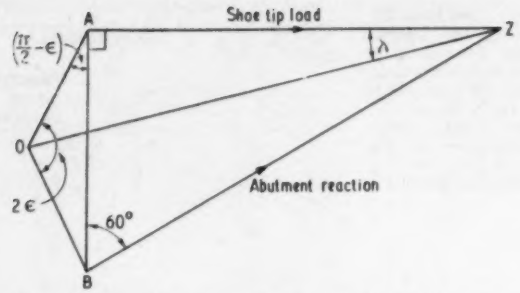


Fig. 10. Diagram for the Girling leading shoe with 60 deg abutments

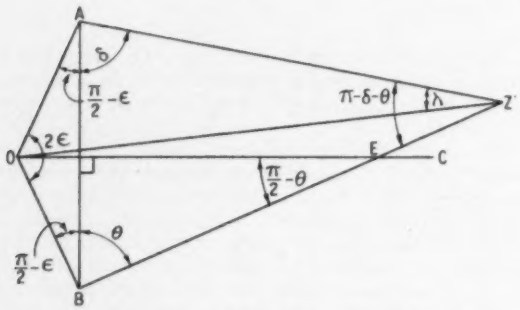


Fig. 11. General case for the determination of the forces and reactions that are obtained with brake shoes having inclined abutments

This is the general equation for F for a pivoted shoe. It may be further simplified by manipulation, and by substituting $2\rho k$ for AB. In making the latter substitution we are assuming that A is the shoe tip and that the shoe tip force is at right angles to AB. This is not necessarily so, and should be verified in each particular case, before using the revised formula.

Substituting as outlined and simplifying;

$$F=\frac{2\rho k}{r\ sec\ \sigma\ cosec\ \beta\ cos\ (\beta+\sigma-\theta)-l} \quad \ldots\ldots\ldots\ldots\ldots\ldots(5)$$

In the special case considered by Waller, OE coincides with OD, so that θ and σ are both zero; also $r=k$. Making these substitutions:

$$F=\frac{2l}{cot\ \beta-l/_k} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(6)$$

Equation (6) is identical with that derived by Waller.

Pivoted shoe—trailing

Equations (4), (5) and (6) refer to a leading shoe. The equations for a trailing shoe can be found in a similar way, and the results are similar in every respect except that the sign between the terms in the denominator is plus instead of minus.

APPENDIX 3

Inclined abutments, Girling and others

In Fig. 10, O represents the shoe centre, A the shoe tip and B the abutment point. At the abutment, the action is at 60 deg instead of at a right-angle to AB as with parallel abutments. The lines of action of the shoe tip load and abutment reaction meet at Z. Join OZ and let $AZO=\lambda$. Z is one of the points that determines the position of the resultant reaction of the drum. The angle AOB is fixed, as is also length OA. Let $\angle AOB=2\epsilon$ and $OA=OB=r\rho$.

It is evident that the whole system is fixed. In any problem of this kind, the first procedure is to find by simple trigonometry all lengths and angles, including angle AZO.

Fig. 11 represents the general case when $\angle BAZ$ is not a right-angle but is angle δ. Also, $\angle ABZ=\theta$, and OC is the centre-line of the shoe, OC bisecting AOB. It can be seen that $\angle BEO=\angle ZEC=\frac{\pi}{2}-\theta$; this is used in the subsequent proof.

As a preliminary, the various lengths and angles must be calculated. The angles of triangle ABZ are known, also length AB can readily be found in terms of length OA and angle ϵ. Hence AZ and BZ can be found by the sine formula. Triangle OAZ can then be completely determined, because two sides and the included angle OAZ are known.

For the derivation of an expression for shoe factor, Fig. 12, the lettering is the same as in Fig. 11, so far as it goes. In addition, OX is the line through the centre of the brake lining, making an angle ϕ with OC, the centre-line; OD is the diameter of the construction circle; and P is the point corresponding to an angle of friction β.

Join PD. Then $\angle ODP=\beta$ and $DOP=\frac{\pi}{2}-\beta$

Join ZP. Then ZP is the line of action of the resultant reaction of the drum

Let ZP make an angle γ with ZA.

The calculation of the shoe factor depends on the determination of the angle γ. It is evident that if γ is known, all the angles making up angle AZB will be known. The expression for shoe factor will include the lengths of the perpendiculars from O and B respectively to the resultant ZP; these lengths can be found in terms of known lengths ZO and ZB, and the appropriate angles.

To find the angle γ:

$$OA=r\rho$$
$$OD=l\rho$$

In $\Delta\ ZAO$, $\dfrac{ZO}{sin\ (\pi/2-\epsilon+\delta)}=\dfrac{r\rho}{sin\ \lambda}$

Hence, $ZO=\dfrac{r\rho\ sin\ (\pi/2-\epsilon+\delta)}{sin\ \lambda}$

From Fig. 12 it can be seen that $\angle OPG$ is $(\pi+\beta-\phi-\delta-\gamma)$.

In $\Delta\ ZOP$,

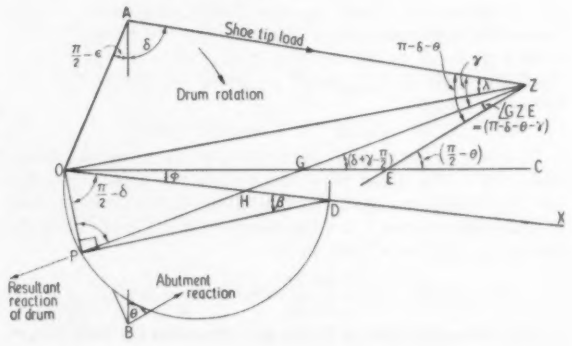
$$\frac{ZO}{sin\ (\pi+\beta-\phi-\delta-\gamma)}=\frac{OP}{sin\ (\gamma-\lambda)}$$
$$=\frac{l\rho\ sin\ \beta}{sin\ (\gamma-\lambda)}$$
$$\therefore\ sin\ (\gamma-\lambda)=\frac{l\rho\ sin\ \beta\ sin\ (\pi+\beta-\phi-\delta-\gamma)\ sin\ \lambda}{r\rho\ sin\ (\pi/2-\epsilon+\delta)}$$

Fig. 12. General case for a leading shoe with inclined abutments

$$\angle OGH=\angle ZGE=\frac{\pi}{2}-\theta-\pi+\delta+\theta+\gamma$$
$$=\delta+\gamma-\frac{\pi}{2}$$

In Δ GOP,

$$\angle OPG=\pi-\frac{\pi}{2}+\beta-\phi-\delta-\gamma+\frac{\pi}{2}$$
$$=\pi+\beta-\phi-\delta-\gamma$$



$$\sin(\gamma-\lambda) = K \sin\beta \sin(\pi+\beta-\phi-\delta-\gamma)$$

Where $K = \dfrac{l \sin\lambda}{r \cos(\epsilon-\delta)}$, which is a constant

To solve for γ, expand $\sin(\gamma-\lambda)$ and $\sin(\pi+\beta-\phi-\delta-\gamma)$, collect terms under $\sin\gamma$, $\cos\gamma$, giving:

$$\tan\gamma = \frac{\sin\lambda + K \sin\beta \sin(\pi+\beta-\phi-\delta)}{\cos\lambda + K \sin\beta \cos(\pi+\beta-\phi-\delta)} = \frac{\sin\lambda - K \sin\beta \sin(\beta-\phi-\delta)}{\cos\lambda - K \sin\beta \cos(\beta-\phi-\delta)}$$

Where $K = \dfrac{l \sin\lambda}{r \cos(\epsilon-\delta)}$ .....(1)

Shoe Factor

Proceeding as before,

Shoe factor, $F = \dfrac{T}{P} = \dfrac{(\text{perpendicular from } O \text{ to } PZ)\ AB}{(\text{perpendicular to } B \text{ to } PZ)\ \rho}$

$$F = \frac{OZ \sin(\gamma-\lambda)\ AB}{BZ \sin(\pi-\delta-\theta-\gamma)\ \rho}$$

This reduces to

$$F = \frac{C \sin(\gamma-\lambda)}{\sin(\pi-\delta-\theta-\gamma)}$$

Where C is a constant $= \dfrac{OZ \times AB}{BZ \times \rho}$ .....(2)

Modification of formula if δ is a right-angle

To find the expression if δ is 90 deg, substitute $\frac{\pi}{2}$ for δ

We have, $K = \dfrac{l \sin\lambda}{r \cos\left(\epsilon-\frac{\pi}{2}\right)} = \dfrac{l \sin\lambda}{r \sin\epsilon}$ .....(3)

In the expression for $\tan\gamma$, $\sin(\beta-\phi-\delta)$ becomes $\sin\left(\beta-\phi-\frac{\pi}{2}\right) = -\sin\left(\frac{\pi}{2}-\beta+\phi\right)$

$$= -\sin\left[\frac{\pi}{2}-(\beta-\phi)\right] = -\cos(\beta-\phi)$$

Similarly, $\cos(\beta-\phi-\delta)$ becomes $\sin(\beta-\phi)$

Hence $\tan\gamma = \dfrac{\sin\lambda + K \sin\beta \cos(\beta-\phi)}{\cos\lambda - K \sin\beta \sin(\beta-\phi)}$ .....(4)

The expression for shoe factor becomes

$$F = \frac{C \sin(\gamma-\lambda)}{\cos(\theta+\gamma)}$$ .....(5)

where C is unchanged, formula 2.

APPENDIX 4

Semi-flexible shoe

Acres discusses the case of a semi-flexible shoe, in which, by definition, the pressure is evenly distributed over the whole surface. It is unlikely that such a condition is ever fully realized in practice, but it is interesting to find what would happen if it were. Firstly, the method of operation of the brake does not come into the calculation: the fact that the pressure is evenly distributed over the whole surface of the linings is sufficient to determine the line of action of the resultant reaction of the drum on the shoe for a given coefficient of friction.

According to Acres, all resultants must pass through a fixed point Z, Fig. 13, and this point is at the intersection of the axis OX through the lining centre and the construction circle. The diameter of the construction circle is $a\rho/\sin\alpha$, thus differing from the expression for a rigid shoe.

In Fig. 13, O is the shoe centre, A the shoe tip, B the abutment point, or pivot, and OC is the line, through the shoe centre, bisecting angle AOB. P is the point on the construction circle, and its position is determined by the angle of friction β; the line of action of the resultant is ZP; also, OP is perpendicular to ZP, and angle $OZP = \beta$.

From B, draw the straight line BG perpendicular to ZP.

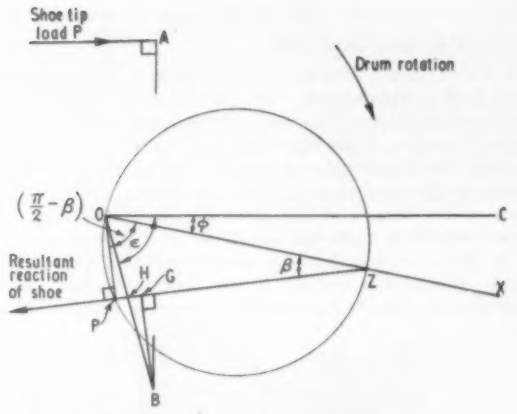


Fig. 13. Diagram for the semi-flexible, leading shoe type layout

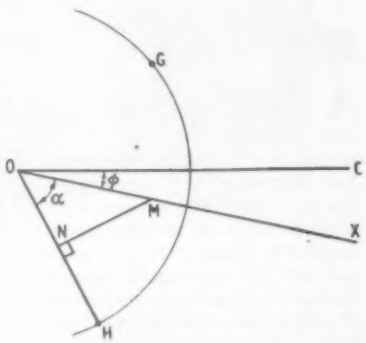


Fig. 14. Diagram for the case of a shoe, only the extreme ends of which contact the drum. This is a condition that might occur after heavy braking over a prolonged period, followed by cooling

Using the method that is fully outlined in Appendix 1:

$$F = \frac{T}{P} = \frac{AB \times OP}{\rho \times BG} = \frac{AB \times OH}{\rho \times BH}$$ .....(1)

To find OH, it is necessary to know the angle POH:

$$\angle POH = \angle POZ - \angle HOZ = \frac{\pi}{2} - \beta - \epsilon + \phi$$

$$\therefore\ OH = OP \sec\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right) = OZ \sin\beta \sec\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right) = l\rho \sin\beta \sec\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right)$$

Also, $BH = BO - OH$

$$= \rho\left[r - l \sin\beta \sec\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right)\right]$$

Also, $AB = 2k\rho$

Substituting in (1):

$$F = \frac{2k\rho\, l\rho \sin\beta \sec\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right)}{\rho\,\rho\left[r - l \sin\beta \sec\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right)\right]} = \frac{2lk}{r \operatorname{cosec}\beta \cos\left(\frac{\pi}{2}-\beta-\epsilon+\phi\right) - l} = \frac{2lk}{r \operatorname{cosec}\beta \sin(\beta+\epsilon-\phi) - l}$$ .....(2)

APPENDIX 5

Contact at liner ends only

Frequent heavy braking over a prolonged period produces high brake temperatures, thus temporarily increasing the radius of the drum. The brake lining wears to this increased radius; however, if during subsequent running, little braking is done, the drum will cool and, of course, contract. Under these conditions it is possible to get contact at the extreme ends of the liner, and this gives an abnormally high shoe factor, which is frequently the cause of snatching.

In Fig. 14, O is the shoe centre, and OC and OX the lines through the centres of the shoe and brake linings respectively. Under the extreme conditions described in the previous paragraph, contact is at points G and H only. Acres shows that the behaviour of the system can be predicted by using a construction circle that passes through O, G, and H.

To determine the diameter of the construction circle, proceed as follows. The centre of the construction circle must be on line OX, since OX obviously bisects GH at right-angles. To fix the centre, bisect OH at right-angles, with line NM cutting OX at M. Then M is the centre of the construction circle. The radius of the construction circle is OM. Now $OH=\rho$, $ON=\rho/2$ and $OM=ON \sec \alpha = (\rho \sec \alpha)/2$. Then l is the ratio of the diameter of the construction circle to the radius ρ of the drum. Hence $l=\sec \alpha$.

New British Cars

SUMMARY OF THE NOTEWORTHY FEATURES OF RECENTLY INTRODUCED VEHICLES

Rolls-Royce and Bentley

A NEW car engine has been introduced by Rolls-Royce Ltd. Of itself, this event is of great importance in the automobile sphere, but the interest is enhanced by the fact that the engine is of the V-eight type. Such a layout has not previously been adopted by the Company, although the V-twelve unit of the late pre-war Phantom III will be recalled by many. The new engine was designed because it was felt that the existing six-cylinder engine could not be further developed without departure from the rigid standards of refinement set by Rolls-Royce. With its inherently good balance and compact dimensions, the V-eight layout offered obvious advantages as a successor.

The new engine is fitted to the latest versions of the Rolls-Royce Silver Cloud and Bentley Series S cars, which are otherwise virtually unchanged. Its outstanding features include over-square cylinder dimensions, hydraulic tappets, an unusual two-carburettor induction system and a multiple vee-belt drive to the accessories. The use of high-silicon aluminium alloy for the cylinder heads and the crankcase and cylinder block casting, in conjunction with wet liners, has resulted in a dry weight no greater than that of the six-cylinder engine.

A swept volume of 6,230 cm^3 is given by the bore and stroke dimensions of 104·4 mm and 91·44 mm respectively; so the stroke : bore ratio is 0·876 : 1. The crankshaft is of chromium-molybdenum steel, and is statically and dynamically balanced. It runs in five main bearings, of the steel-backed type, with copper-lead linings coated with indium. On each crankpin, two connecting rods are disposed one in front of the other, unlike those of the famous Rolls-Royce aircraft engines which were of the forked-and-plain type. In consequence, the bores of the right-hand cylinder block are staggered ahead of those of the left-hand block. Copper-lead alloy is employed also for the big-end bearings. Longitudinal location of the crankshaft is effected at the centre main bearing, on each side of which is a thrust washer.

The camshaft is of Monikrom cast iron and occupies the orthodox position, in the 90 deg angle between the cylinder banks. A helical, steel gear on the crankshaft drives the camshaft gear, which is of large diameter and unusual construction. For quietness of operation, it is a casting of light alloy and has a web of radially corrugated disc form between the hub and the flange on which the teeth are cut. The corrugations provide the necessary rigidity with the minimum of weight, which is said to be no more than 1 lb for the complete gear. At the rear of the camshaft is a spiral gear driving the vertically mounted distributor.

As is customary in the design of hydraulic tappets, use is made of the oil pump pressure to keep the tappets topped-up, and any overflow due to aeration passes up the hollow push rods to lubricate the ball ends of the rockers. The push rods are of Bundy tubing, with inserted steel ends, and they actuate the valves by means of an orthodox system of overhead rockers. All the valves of each bank are in line, but the plane of the valve stem axes is inclined inward at 28 deg to the plane of the cylinder axes. The combustion chambers

Externally, the new Rolls-Royce Silver Cloud II is identical with its predecessor. The proportions are good, and the lines have the elegance expected of this make

are of approximately wedge shape in cross section, but are of bath tub form as viewed in plan. Since the design aim has been a wide spread of torque rather than high maximum b.h.p., the valves are relatively small in comparison with the bore. They seat on inserts of austenitic steel and are closed by single helical springs.

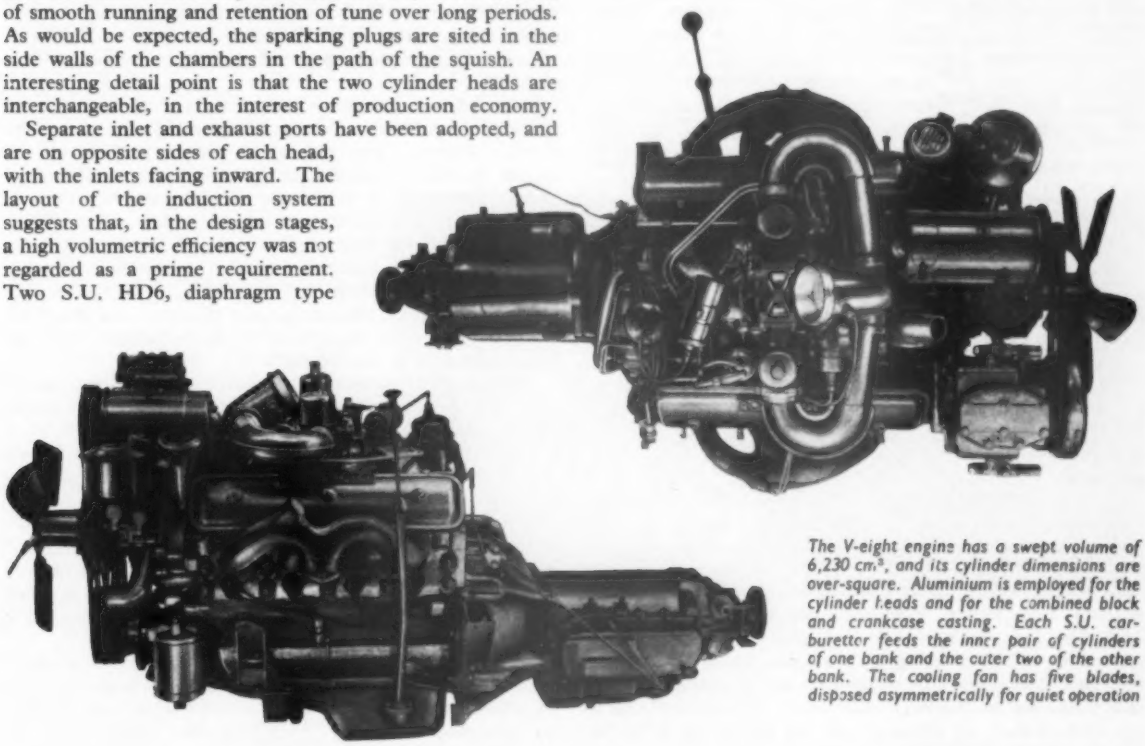
In each piston crown is a central, toroidal depression; by this means, the ratio of exposed surface : volume of the combustion chamber is kept to a minimum. The compression ratio has been deliberately restricted to 8 : 1, in the interest of smooth running and retention of tune over long periods. As would be expected, the sparking plugs are sited in the side walls of the chambers in the path of the squish. An interesting detail point is that the two cylinder heads are interchangeable, in the interest of production economy.

Separate inlet and exhaust ports have been adopted, and are on opposite sides of each head, with the inlets facing inward. The layout of the induction system suggests that, in the design stages, a high volumetric efficiency was not regarded as a prime requirement. Two S.U. HD6, diaphragm type horizontal carburettors, of 1¾ in bore, are bolted to a two-piece manifold. The upper portion merely turns the flow from each carburettor downward through a right-angle into the divided main portion of the manifold. One carburettor feeds the inner pair of cylinders of one bank and the outer pair of the second bank, while with the other the arrangement is mirrored; there is no balance connection between the carburettors.

This system of manifolding ensures that the depression impulses on each carburettor are evenly spaced. Also, the passages to the various cylinders are all of similar lengths, while any unbalance that may arise between the carburettors is shared between the banks of cylinders. An automatic choke is embodied in the system, and the impregnated-paper air filter is attached to the bonnet lid, and lifts with it to improve the accessibility of the engine.

The oil pump has helical gears, and is driven by a spiral gear on the crankshaft, ahead of the camshaft driving pinion. As would be expected, the lubrication system is very thorough, and the sump capacity is 12 pints. A torsional vibration damper is fitted to the leading end of the crankshaft, and in front of it is the four-groove pulley that carries the vee-belts driving the tandem fan and water pump, the generator, the oil pump for the powered steering and, if fitted, the compressor for the optional refrigeration system. The complete power unit is suspended on rubber at three points. To attain the traditional level of exhaust quietness, there are three acoustic type silencers in series, each tuned to absorb a different range of frequencies. In accordance with their custom, the manufacturers are not releasing any performance figures for the engine.

No changes have been found necessary to the automatic transmission, which has been standard on Rolls-Royce and Bentley cars for several years. It is based on an earlier General Motors design and, unlike many current systems, does not embody a torque converter, with its attendant low efficiency at the larger torque multiplications. Instead, there is a fluid coupling used in conjunction with a four-speed gearbox of epicyclic type. The forward ratios are 1 : 1, 1·45 : 1, 2·63 : 1 and 3·92 : 1, while reverse is 4·33 : 1. A two-piece propeller shaft transmits the drive to the semi-floating rear axle, which has a hypoid-bevel reduction of 3·08 : 1.



The V-eight engine has a swept volume of 6,230 cm³, and its cylinder dimensions are over-square. Aluminium is employed for the cylinder heads and for the combined block and crankcase casting. Each S.U. carburettor feeds the inner pair of cylinders of one bank and the outer two of the other bank. The cooling fan has five blades, disposed asymmetrically for quiet operation

The rear suspension layout comprises semi-elliptic springs and a radius arm, at the right-hand side, between the axle and the frame, to limit wind-up. An interesting feature of this suspension system is the electrical adjustment of the setting of the lever-type dampers, effected by a switch on the steering column. The spring rate is 90 lb/in and the periodicity is 75 c/min. A roll centre height of 14 in is quoted by the manufacturers, and the roll stiffness is 320 lb-ft/deg.

At the front, a suspension layout of double wishbones and coil springs is used. The wishbones are of unequal length and are mounted at a trailing angle of 22½ deg to the axis of the front wheels. Lever type dampers control the action of the springs, and an anti-roll bar is fitted. The figures for spring rate, periodicity, roll centre height and roll stiffness are respectively 90 lb/in, measured at the wheel, 55 c/min, ⅞ in and 480 lb-ft/deg.

A three-piece track rod and two slave levers are employed in the steering linkage. The cam-and-roller steering box has a ratio of 18·7 : 1, and 4¼ turns of the steering wheel are

Rolls-Royce Silver Cloud II and Bentley S2

SPECIFICATION DATA

Engine

Number of cylinders 8, 90 deg V lay-out
Bore 104·4 mm
Stroke 91·44 mm
Swept volume 6,230 cm³
Compression ratio 8 : 1
Crankshaft Five-bearing, forged steel, statically and dynamically balanced
Valves Overhead, inclined, push rod operated from single camshaft, through hydraulic tappets
Combustion chamber type Wedge in conjunction with toroidal depression in piston crown
Carburettors Two S.U. HD6, horizontal
Fuel pumps Two S.U. electric, independent
Dry weight 681 lb, excluding fluid coupling

Transmission

Make and type Rolls-Royce automatic, incorporating fluid coupling and four-speed epicyclic gearbox
Gear ratios

fourth	1 : 1
third	1·45 : 1
second	2·63 : 1
first	3·92 : 1
reverse	4·33 : 1

Dry weight 209 lb, including coupling
Propeller shaft Open, two-piece, with needle roller bearing universal joints; medial universal joint flexibly mounted

Rear axle

Type Semi-floating, with hypoid-bevel reduction
Ratio 3·08 : 1

Suspension

Front Double semi-trailing wishbones and coil springs, with lever type dampers and anti-roll bar
Rear Semi-elliptic springs, with single radius arm and lever type dampers, adjustable electrically

Steering

Make and type Rolls-Royce cam-and-roller, with hydraulic servo assistance
Ratio 18·7 : 1
Turning circle 41 ft 8 in
Turns from lock to lock 4¼

Brakes

Front Rolls-Royce-Girling hydraulic, with two trailing shoes and mechanical servo assistance. Drum diameter, 11¼ in; shoe width, 3 in
Rear Rolls-Royce-Girling hydraulic and mechanical, with leading and trailing shoes and mechanical servo assistance. Drum diameter, 11¼ in; shoe width, 3 in. Mechanical actuation by handbrake
Distribution of braking effort
front 62·4 per cent
rear 37·6 per cent

Wheels and tyres

Wheel type Pressed-steel disc, with five-stud attachment and covering discs
Tyre size 8·20—15 in
Pressures, saloons front 22 lb in²
rear 27 lb in²

Dimensions

Wheelbase 11 ft 2 in (11 ft 6 in for long-wheelbase Silver Cloud)
Front track 4 ft 10 in
Rear track 5 ft 0 in
Ground clearance 7 in
Overall length 17 ft 7¾ in
Overall width 6 ft 2¾ in
Overall height, saloons 5 ft 4¼ in
Kerb weight with full fuel tank, saloons 4,579 lb
Frontal area, saloons 26·4 ft²

required from lock to lock; for a car with a wheelbase of 11 ft 2 in, the turning circle of 41 ft 8 in is good. Power assistance of the steering is provided, by means of the hydraulic pump already mentioned and a single external jack actuating the left-hand slave lever. The control valves are embodied in the steering box, and are operated by the initial movement of the worm member.

As a safety measure, there are two independent hydraulic systems for the brakes, each with its own master cylinder. The brakes are made by Girling, to Rolls-Royce specification, and have drums of 11¼ in diameter, with a shoe width of 3 in, providing a total lining area of 240 in², or 117 in²/ton kerb weight. Two trailing shoes are employed in the front brakes, which have purely hydraulic actuation. The rear brakes, on the other hand, have leading and trailing shoes, and a composite operating system, whereby 70 per cent of the effort is applied hydraulically and the rest mechanically. As for many years, a mechanical servo, effected by a disc-type friction clutch driven from the output end of the gearbox, is employed. A modification to the latest cars is an increase in the proportion of the braking effort applied to the front wheels. On the previous models it was 57·7 per cent, and it is now 62·4 per cent.

The chassis frame has the necessary beam and torsional stiffness to form a satisfactory platform for a large body. It has box-section side members and cross members, and all the joints are welded. The standard closed body for both the Rolls-Royce Silver Cloud II and the Bentley Series S is a handsome saloon, providing comfortable accommodation for up to six people; a drop-head coupé, with coachwork by H. J. Mulliner, is available in either make. Additionally, there is a Rolls-Royce chassis of longer, 11 ft 6 in, wheelbase, on which is fitted a saloon with a division.

Included in the lavish equipment of these cars are electrically operated windows, all of which are under the driver's control. The redesigned heating and ventilating system is very comprehensive, and includes a two-speed blower fan and an electrically heated rear window. If desired, full air conditioning can be incorporated. When it is specified, the refrigeration unit, of the vapour compression cycle type, is installed within the off-side front wing, and additional insulation of the roof and the exhaust system is provided. Since the refrigerator receives both the fresh and recirculated air, rapid lowering of the interior temperature is obtained in hot weather.

The standard saloon has a ratio of wheelbase : mean track of 2·09 : 1, and the ratios of wheelbase : overall length and of mean track : overall width are respectively 0·59 : 1 and 0·79 : 1. According to the manufacturers, the frontal area is 26·4 ft². Of the kerb weight, a little over 2 tons, 48·5 per cent is carried on the front wheels.

Replacing the Rolls-Royce Silver Wraith limousine is the Phantom V, the largest car ever made by the Company. As was the Silver Wraith, it is available as a chassis or with bodies by specialist coachbuilders. The new car is fitted with the V-eight engine and automatic transmission, but it has a lower rear axle ratio of 3·89 : 1.

In general, the chassis details are similar to those of the Silver Cloud II and Bentley Series S. However, since the wheelbase is no less than 12 ft, and the track is wider than on the models already described, the chassis frame is strengthened and embodies a cruciform bracing. Also, the layout of the rear suspension differs in that the springs are of asymmetric type, and the tyre size is increased from 8·20—15 in to 8·90—15 in. The front track is 5 ft 0⅞ in, the rear track is 5 ft 4 in and the overall length is 19 ft 10 in. Even with such dimensions, the turning circle has been kept down to the moderate figure of 48 ft 9 in.

The bodies fitted to the Phantom V are equipped in the height of lavishness. One of the first examples will be displayed at the Paris Show, which opens on the 1st October, and has limousine coachwork by James Young. Other bodies to be fitted include a seven-passenger limousine by Park Ward, and a touring limousine by H. J. Mulliner; both will be exhibited at the Earls Court Show in October.

Rootes Models

CREDIT for the introduction of the first quantity-produced, British light cars with fully automatic transmission is shared by the Rootes group and S. Smith & Sons (England) Ltd. The latest Hillman Minx Series IIIA and Singer Gazelle models are to be optionally available in what is termed the Easidrive form, in which they are equipped with the Smiths transmission, a three-speed unit incorporating two magnetic-particle couplings. Other changes to these cars include increased engine power output, modified styling and improved visibility. In standard form, that is, without the Easidrive transmission, the vehicles now have the same gearbox as the Sunbeam Alpine described in last month's issue of *Automobile Engineer*; this gearbox has a more satisfactory spacing of the ratios than did its predecessor and, on the Minx and Gazelle, is offered with the option of floor-mounted or steering-column shift lever.

The third of the Rootes trio of new cars is the Sunbeam Rapier Mark III. This also has a more powerful engine, which in most respects resembles that of the Alpine model. In addition, the later conventional gearbox is installed, and is controlled by a floor-mounted lever; because of its more sporting nature, this car is not available with the automatic transmission. Lockheed disc brakes are now fitted on the front wheels, but drum type brakes are retained for the rear wheels.

There is a considerable degree of rationalization between these three ranges of cars. All have the same basic 1,494 cm^3 four-cylinder engine, which has a bore of 79·0 mm and a stroke of 76·2 mm, giving a stroke : bore ratio of 0·965 : 1. The crankshaft runs in three white-metal lined main bearings, and the big-end bearings are lined with copper-lead, with an indium coating; the ratio of connecting-rod length : stroke is 1·92 : 1. Combustion chambers of bath tub form are employed, but the minor, or transverse axis is slightly displaced longitudinally from the transverse axis of the bore. The in-line, overhead valves are actuated by push rods from a chain-driven side-mounted camshaft.

The versions of the engines installed in the Minx Series IIIA and the Gazelle have cylinder heads of cast iron, with vertical valves, and a compression ratio of 8·5 : 1. A Zenith VIG downdraught carburettor is now fitted to the Minx, and the combination of this and improvements in the manifolding has resulted in an increase of power output from the 52·5 b.h.p. at 4,400 r.p.m. of the earlier unit to 56·5 b.h.p. at 4,600 r.p.m. Probably of greater value in normal motoring, however, is the increase in the maximum gross torque from 78 lb-ft to 83 lb-ft, accompanied by a small decrease, from 2,100 to 2,000 r.p.m., in the speed at which it occurs. The new carburettor is claimed to give improved economy of operation, and certainly the minimum brake specific fuel consumption of 0·50 to 0·51 pt/b.h.p-hr is satisfactory for such a unit.

This engine has a ratio of maximum torque : torque at maximum b.h.p. of 1·28 : 1, and the ratio of speed at maximum torque : speed at maximum b.h.p. is 0·435 : 1. In terms of b.h.p/litre and b.h.p/in^2 piston area, the power output figures are respectively 37·8 and 1·87. The power-weight figures, expressed as b.h.p/lb engine weight and b.h.p/ton kerb weight of the de luxe saloon, are 0·165 and 56·6. At maximum b.h.p., the mean piston speed is 2,300 ft/min.

In the case of the Gazelle engine, the inlet and exhaust manifold of last year's Rapier engine have been adopted, and two Solex 32 PB1S downdraught carburettors replace the previous single carburettor of that make. As a result, the gross power output has been increased from 60 b.h.p. at 4,500 r.p.m. to 64 b.h.p. at 4,600 r.p.m. On the other hand, the maximum torque is virtually unaltered: it was previously quoted as 82·8 lb-ft at 2,300 r.p.m. and is now 82·6 lb-ft at 2,400 to 2,800 r.p.m.

The performance criteria for this engine are: mean piston speed, 2,300 ft/min; ratio of maximum torque : torque at maximum b.h.p., 1·128 : 1; ratio of speed at maximum torque : speed at maximum b.h.p., 0·522 : 1; minimum brake specific fuel consumption, 0·50 to 0·52 pt/b.h.p-hr; 42·8 b.h.p/litre, 2·11 b.h.p/in^2 piston area, 0·186 b.h.p/lb engine weight, and 61·9 b.h.p/ton vehicle kerb weight for the saloon.

On the latest Rapier engine, the aluminium cylinder head

Right: In this view, the modified radiator grille of the Hillman Minx Series IIIA cars can be seen. The engine develops more power than before, the windscreen is deeper, and shallow tail fins are now embodied

Left: The latest Singer Gazelle also has a more powerful engine and increased area of the windscreen. To suit the changes at the rear, notably the incorporation of fins, the side flashes are of a modified shape

of the Alpine unit is employed. It may be recalled that this has the valves inclined at 4½ deg to the vertical; the compression ratio is 9·2 : 1, and the inlet manifold is water jacketed because a hot spot was impracticable with the Alpine's free-flow exhaust system. The Rapier has a cast iron exhaust manifold which, although slightly less efficient than the fabricated tubular assembly of the Alpine, has good flow characteristics. Also, the Zenith downdraught carburettors are of the 36 WIA2 type, instead of 36 WIP2, and have a common oil bath air filter in place of individual filters.

As a result of these differences, the performance is naturally rather lower than that of the Alpine engine, though it is considerably better than that of the previous Rapier unit. The gross power output is 78 b.h.p. at 5,400 r.p.m., as against the 83·5 b.h.p. of the Alpine at the same speed and the 73 b.h.p. at 5,200 r.p.m. of the model's predecessor. There has been a small increase in the maximum torque, from 81·3 lb-ft to 83 lb-ft, but the speed at which this is attained has also gone up, from 3,000 r.p.m. to 3,500 r.p.m.

For comparison with the corresponding figures quoted for the other engines, the mean piston speed is 2,700 ft/min, and the ratio of maximum torque : torque at maximum b.h.p. is 1·093 : 1, while that of speed at maximum torque : speed at maximum b.h.p. is 0·648 : 1. The b.h.p/litre and b.h.p/in² piston area are respectively 52·2 and 2·58; the b.h.p/lb engine weight and the b.h.p/ton kerb weight are 0·235 and 74·7.

With the four-speed transmission, a Borg and Beck single-dry-plate clutch, which has a diameter of 8 in and a friction area of 2×24·3 in², is installed in the bell-housing. The gearbox has synchromesh on second, third and fourth gears, and provides ratios of 1 : 1, 1·392 : 1, 2·141 : 1 and 3·346 : 1, with a reverse of 4·239 : 1. To conform with the more common practice abroad, the gate layout of the steering column gear change, which is still available on the Minx and Gazelle, has been reversed: the plane of movement for first and second gears is now above that for third and top instead of being below. The floor-mounted lever, optional on these two models and standard on the Rapier, is short and conveniently placed; it actuates the selector mechanism through a remote control similar to that of the Alpine model.

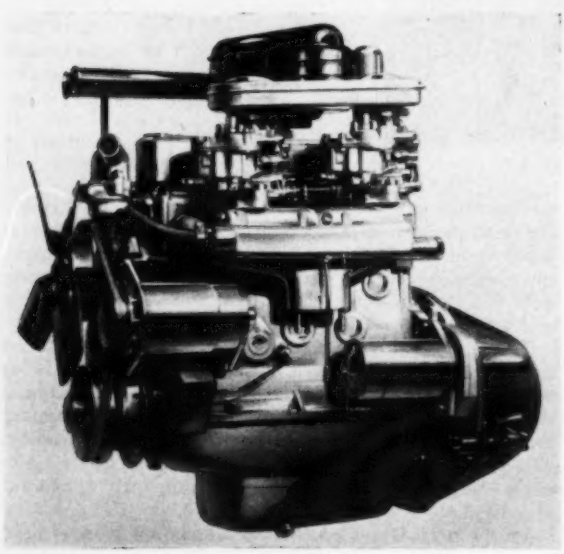
A detailed description of the Smiths automatic transmission was given in the September 1957 issue of *Automobile Engineer*. The system is essentially simple and robust, and, since it does not incorporate a torque convertor, its efficiency over the whole range, including the intermediate gears, is high. It is therefore fundamentally better suited for use with engines of less than 2 litres swept volume than are the more usual types of automatic transmission, with hydrodynamic drives.

The principle of the Smiths magnetic-particle coupling is by now too well known to need more than brief mention here. There are three co-axial members, the outer a stationary field member, of annular form, into which is built the excitation coil. This field unit surrounds the revolving input member, and there is an air gap between the two. The innermost portion of the coupling is the output member, and the space between its periphery and the input member contains the chromium-iron magnetic particles. When current is passed through the coil, the resultant magnetic field causes the powder to orientate itself between the input and output members in such a way that torque is transmitted from one to the other. The torque capacity is almost directly proportional to the applied current, regulation of which results in a smooth take-up of the drive.

In the automatic transmission application, there is a common field member for both couplings, and a common input member that forms part of the engine flywheel. The output members are installed coaxially in tandem, that in front being for direct drive and the other for indirect drive;

The tail fins of the Minx and Gazelle flare outwards almost horizontally

Below: Features of the Sunbeam Rapier engine are an aluminium cylinder head, with inclined valves, and a water heated inlet manifold

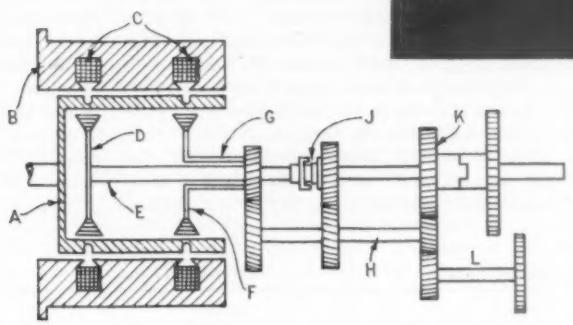


the indirect drive member is mounted on a sleeve round the shaft of the direct drive member. Accompanying this review is a schematic illustration of the couplings and the gearbox. From this it can be seen that the output member of the direct drive coupling is permanently connected to the mainshaft of the gearbox, and so to the propeller shaft; in consequence, it is always revolving at a speed proportional to that of the car. The drive from the output member of the indirect coupling is transmitted through constant-mesh, helical gears to the layshaft. Each output member is resiliently mounted on its hub, coil springs being used to provide a shock-absorbent drive.

On the mainshaft and layshaft are two additional pairs of constant-mesh gears, also of the helical type. The middle pair are for second gear and the rear pair for bottom gear. Engagement is effected by dog clutches in each case; incorporated in the second gear dog member is a baulking device, and a free-wheel is embodied in the mainshaft gear of the first gear pair. This free-wheel not only improves the behaviour in dense traffic, by eliminating the powerful

Right: Apart from the deeper screen, there is little to distinguish the new Rapier from its predecessor. It has a more powerful engine, and disc type front brakes

Below: A diagrammatic representation of the Smiths automatic transmission, available at extra charge on the latest Minx and Gazelle models. It occupies little more space than the orthodox transmission



A *input member;* **B** *stationary field member;* **C** *coils;* **D** *output member, direct drive;* **E** *output shaft, direct drive;* **F** *output member, indirect drive;* **G** *output sleeve, indirect drive;* **H** *layshaft;* **J** *constant-mesh second gear, with dog engagement;* **K** *constant-mesh first gear, with dog engagement and free-wheel;* **L** *reverse idler*

engine braking effect in first gear, but has a secondary purpose that will be explained later.

First gear is engaged mechanically by means of the dog clutch and at the same time the indirect coupling is energized. The drive is then transmitted through the first and third pairs of gears, and the output member of the direct coupling is idling. In second gear, the indirect coupling is still energized, and the second gear dog clutch is engaged by a solenoid. It follows that the drive is taken through the leading two pairs of gears in the box, and the output member of the direct coupling continues to idle, though at a reduced speed relative to the input member. The first gear dogs are not disengaged because the free-wheel permits the mainshaft gear to be over-run.

When top gear is engaged, the second gear solenoid is de-energized, the dogs are disengaged under spring pressure, and the field current is switched from the indirect to the direct coupling. The bottom gear dogs remain in engagement, and the layshaft idles by virtue of the free-wheel. Reverse is obtained manually by the engagement of a straight-tooth gear, sliding on splines on the mainshaft, with another on the third shaft, which is driven by the layshaft first gear pinion in the usual way. In the reverse drive condition, the indirect coupling is energized and both the first and second gear dogs are disengaged.

Controlling the transmission is a governor unit, which is driven by a spiral gear on the tail of the mainshaft, and is sensitive both to road speed and to throttle opening. Its signals are transmitted to a control box, containing appropriate electrical relays and resistances to effect the gear changes. In the governor, use is made of hysteresis effects to cause upward changes to occur at higher road speeds than those at which the corresponding downward changes are made, thereby avoiding hunting between the ratios at a nearly constant road speed. Additionally, the governor prevents the engagement of second from top gear above a certain road speed, to avoid the possibility of engine damage through over-speeding.

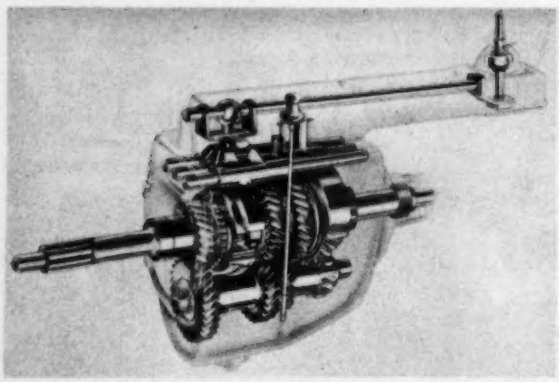
A selector lever is mounted on the steering column and has an indicator quadrant with four positions: R, N, L and D, indicating reverse, neutral, low and drive. The engine can be started only when the lever is in the neutral position. With the lever in the drive setting, all changes are made automatically, except that engagement of the next lower gear can be effected, below the maximum speed for that gear, by depressing the throttle pedal. Selection of L results in the second gear signal being continuously transmitted, so that this gear is engaged and held at all speeds below its maximum. Reverse gear can hardly be engaged accidentally because of the necessity of moving the lever into a different plane of travel. The ratios of the Smiths transmission are 1 : 1, 1·590 : 1 and 2·963 : 1, with a reverse of 3·076 : 1.

With either type of transmission, the propeller shaft is of the conventional open type, and a semi-floating rear axle, with spiral-bevel reduction, is specified. The normal ratio is 4·55 : 1 but the alternative of 4·78 : 1 is fitted to the Minx and Gazelle Estate Cars with standard transmission, and to all Gazelle and Rapier models if ordered with overdrive. This is of Laycock-de Normanville manufacture and provides a step-up of 0·803 : 1 on fourth and third gears. A road speed in top gear of 15·9 m.p.h/1,000 r.p.m., is given by the 4·55 : 1 axle ratio; the figures for the other ratios can be calculated from the information in the specification data table.

Semi-elliptic springs and telescopic dampers are employed for the rear suspension. At the front, the layout comprises double wishbone links and coil springs; the wishbones are pivoted at an appreciably trailing angle, as was mentioned in the description of the Sunbeam Alpine. The telescopic dampers are mounted co-axially within the coil springs, and the ends of the anti-roll bar are connected to the front arms of the lower wishbones.

The rate of the rear springs on all three makes is 143 lb/in and the periodicity is 73·8 c/min. A roll centre height of 9·23 in and a roll stiffness of 2,877 lb-in/deg are quoted by the manufacturers. For the front suspension, the spring rate, measured at the wheel, is 94·3 lb/in on the Minx and Gazelle, and 114 lb/in on the Rapier. The periodicity is 68·7 c/min, and the roll centre is at ground level; the roll stiffness is 2,217 lb-in/deg.

A double-ended drop arm is fitted to the Burman recirculating-ball type steering box, which is mounted on the dash bulkhead. The track rod is of the three-piece type, and its middle portion passes behind the cylinder block to link the rear end of the drop arm to the rear end of the double-ended slave lever. To the front ends of the drop arm and slave lever are connected the outboard portions of the track rod



The gearbox now fitted as standard to all the 1¼-litre Rootes cars has a remote control, and the spacing of the gear ratios has been improved

assembly, which in turn are coupled to the trailing steering arms. The turning circle is 36 ft and movement of the front wheels from lock to lock involves 3·15 turns of the wheel.

Previously, the Minx had brakes of 8 in diameter, those of the Gazelle were of 9 in diameter, and the Rapier had 9 in drums at the rear and 10 in at the front. All except one of the Series IIIA Minx models have been brought up to the Gazelle specification, the exception being the lower priced Special Saloon, which retains the smaller brakes. The larger units have a shoe width of 1¾ in and a total lining area of 121 in². This area is over 30 per cent greater than that of the 8 in diameter brakes, which have a shoe width of 1¼ in. As hitherto, the brakes are of Lockheed manufacture; those at the front have two leading shoes, whereas a leading-and-trailing shoe layout is used at the rear. Of the total braking effort of the Minx and Gazelle models, 64 per cent is applied to the front wheels.

As has already been indicated, the new Rapier augments still further the ranks of fast cars having disc brakes at the front. Unlike the Alpine, which has Girling equipment, the Rapier has Lockheed brakes, with discs of no less than 10 $\frac{13}{16}$ in diameter. The Rapier is about 175 lb heavier than the Alpine and has more weight on the front wheels, so larger discs were thought advisable to ensure ample braking performance without the need of servo assistance. A thickness of 0·35 in is quoted for the discs, which are of cast iron, and the friction area of each of the two pads in each caliper unit is 5 in². The rear brakes are identical with the 9 in units already described, and the distribution of braking effort between the front and rear wheels is 69 : 31. It has been necessary to widen the front track by ¾ in to accommodate the disc brakes, and the change is claimed to have improved the stability slightly.

No basic alterations have been made to the unitary body structures. The underbody is common to all three makes, and a feature of it is the fabricated, box-section frontal cross member. This carries the front of the power unit and its ends are swept rearwards to form mountings for the pivots of the suspension wishbones and for the upper ends of the coil springs and dampers. The wheel arches and scuttle are integral with the underbody, to which the body sections are welded.

There are three body styles for the Minx and Gazelle, of which the major pressings are common to both makes. One is a four-door saloon, the second is a two-door drop-head coupé and the third is the Estate Car, or station wagon. Two Rapier models are listed, a two-door saloon and a drop-head coupé; these are similar below the waistline, both to each other and to the coupés of the other two makes.

The most noticeable changes from last year's corresponding Minx and Gazelle models are the adoption of a wider and deeper windscreen, and of shallow, almost horizontal tail fins, with an outswept arcuate section, the trailing edge of which forms a partial cowl over the rear lamp groups. While the windscreen modification applies to all models, the tail fins are embodied only on the saloons and convertibles; the lines of the station wagons do not lend themselves to this form of rear-end decoration. The door pillars of the saloons are slimmer than before. On the Rapier, the tail fins are unaltered from those of the previous model. Though again of outswept arcuate section, they are higher than those of the Minx and Gazelle, and project slightly above the waistline. The larger windscreen is also featured on these cars.

As can be seen from the illustrations, the frontal treatment of the three makes remains quite different. The long, horizontal grille of the Minx has been altered: it is now divided into two by a horizontally barred centre section, and its extremities are taken a small way round the front quarters. The Gazelle retains its shield-shape main grille, flanked near the bottom by barred, horizontal intakes, the grilles of which embrace the side lamps. On the Rapier, the main grille is rectangular, with vertical slats, and it also is flanked by horizontally barred, secondary intakes for cooling purposes.

Rootes Models — SPECIFICATION DATA

Engine

Number of cylinders 4
Bore 79·0 mm
Stroke 76·2 mm
Swept volume 1,494 cm³
Maximum b.h.p. (gross) Minx, 56·5 at 4,600 r.p.m.; Gazelle, 64 at 4,600 r.p.m.; Rapier, 78 at 5,400 r.p.m.
Maximum b.m.e.p. Minx, 137 lb/in²; Gazelle, 136·5 lb/in²; Rapier, 137 lb/in²
Maximum torque Minx, 83 lb-ft at 2,000 r.p.m.; Gazelle, 82·6 lb-ft at 2,400 to 2,800 r.p.m.; Rapier 83 lb-ft at 3,500 r.p.m.
Compression ratio Minx and Gazelle, 8·5 : 1; Rapier, 9·2 : 1
Crankshaft Three-bearing, forged steel
Valves Overhead, push rod operated; vertical on Minx and Gazelle, slightly inclined on Rapier
Combustion chamber type Modified bath tub
Carburettors Downdraught. Minx, single Zenith VIG; Gazelle, two Solex 32 PB1S; Rapier, two Zenith 36 WIA2
Fuel pump AC mechanical
Dry weight, including clutch Minx, 343 lb; Gazelle, 350 lb; Rapier, 332 lb

Transmission

Clutch Borg and Beck single dry plate, 8 in diameter, with standard transmission only
Gearbox type Four-speed, with synchromesh on second, third and fourth. Laycock-de Normanville overdrive optional on Gazelle and Rapier. Alternative on Minx and Gazelle of Smiths automatic three-speed transmission, embodying two magnetic-particle couplings
Gear ratios, standard

	Normal	Overdrive
fourth	1 : 1	0·803 : 1
third	1·392 : 1	1·117 : 1
second	2·141 : 1	
first	3·346 : 1	
reverse	4·239 : 1	

Gear ratios, automatic

third	1 : 1
second	1·590 : 1
first	2·963 : 1
reverse	3·076 : 1

Dry weight, including bell-housing
standard 71 lb
standard with overdrive 91 lb
automatic 178 lb
Propeller shaft Open, with needle roller bearing universal joints

Rear axle

Type Semi-floating, with spiral-bevel reduction
Ratio
standard 4·55 : 1
standard with overdrive 4·78 : 1
automatic 4·55 : 1
(Estate Cars have 4·78 : 1 except with automatic transmission)

Suspension

Front Double wishbones and coil springs, with telescopic dampers and anti-roll bar
Rear Semi-elliptic springs, with telescopic dampers

Steering

Make and type Burman recirculating-ball
Ratio 14·5 : 1
Turning circle 36 ft
Turns from lock to lock 3·15

Brakes

Front Lockheed hydraulic. Minx and Gazelle, drum type, with 9 in diameter drums and two leading shoes, 1¾ in wide (8 in diameter and 1¼ in wide on Minx Special Saloon); Rapier, disc type, with 10 $\frac{13}{16}$ in diameter discs
Rear Lockheed hydraulic, drum type, with 9 in diameter drums and leading-and-trailing shoes, 1¾ in wide (8 in diameter and 1¼ in wide on Minx Special Saloon)
Distribution of braking effort
Minx and Gazelle
front 64 per cent
rear 36 per cent
Rapier
front 69 per cent
rear 31 per cent

Wheels and tyres

Wheel type Pressed-steel disc, with four-stud attachment
Tyre size 5·60—15 in (5·50/5.90—15 in on Estate Cars)
Pressures, normally laden
front 24 lb/in²
rear 24 lb/in²

Dimensions, saloons

Wheelbase 8 ft
Track, front Minx and Gazelle, 4 ft 1 in; Rapier, 4 ft 1¾ in
Track, rear 4 ft 0½ in
Ground clearance Minx and Gazelle, 7 in; Rapier, 5¾ in
Overall length Minx, 13 ft 6 in; Gazelle, 13 ft 7½ in; Rapier, 13 ft 6½ in
Overall width Minx and Gazelle, 5 ft 0¾ in; Rapier 5 ft 1 in
Overall height Minx and Gazelle, 4 ft 11¼ in; Rapier, 4 ft 10½ in
Kerb weight, with full fuel tank Minx, 2,237 lb; Gazelle, 2,316 lb; Rapier 2,340 lb

The remaining variations between the makes are mainly in respect of the decoration and trim. On the Minx models, the previous single full-length bright moulding on the body sides is replaced by a narrow double moulding, while the Gazelle still has the flared double moulding, though slightly restyled to suit the altered lines of the tail. The side mouldings of the Rapier, too, have been modified to give a rather lower appearance. Internally, the Gazelle and Rapier, as befits their higher price, are more luxuriously fitted out, and the trim includes burr walnut for the facia panel. The bench type front seat of the Minx models has been lowered slightly, its runners moved further back, and its contours altered to enhance the comfort. On the Rapier, also, comfort has been improved by the adoption of higher squabs to the separate front seats.

In the accompanying table of specification data, the major dimensions are given for the three ranges of cars. The overall lengths differ, because of the use of bumpers of varying types, and the exhaust system of the Rapier gives a lower ground clearance than do those of the other two makes. The kerb weight of the Minx is rather less than that of the more lavishly equipped Gazelle or Rapier models. In the case of the Minx and Gazelle, the ratio of wheelbase : mean track is 1·97 : 1, and that of mean track : overall width is 0·803 : 1, the corresponding figures for the Rapier are 1·95 : 1 and 0·805 : 1. The ratio of the wheelbase : overall length is 0·593 : 1 for the Minx, 0·587 : 1 for the Gazelle and 0·591 : 1 for the Rapier. For the saloons of the three makes, the front : rear kerb weight distributions are respectively 55 : 45, 54 : 46 and 54 : 46.

Finally, the popular Hillman Husky utility model continues in production, but with modifications to the engine. This is basically the earlier Minx overhead valve unit, with a swept volume of 1,390 cm³ and the bore and stroke equal at 76·3 mm. The compression ratio has been raised from 7·0 : 1 to 8·0 : 1, and a Zenith 30VIG carburettor is now fitted. As a result, the gross power output has risen from 43 b.h.p. at 4,000 r.p.m. to 51 b.h.p. at 4,400 r.p.m., and the maximum torque is now 72 lb-ft at 2,200 r.p.m., as against the previous 66·4 lb-ft at 1,600 r.p.m. Not only is the all-round performance of the vehicle markedly improved by the changes, but the fuel economy under normal running conditions is said to be appreciably better.

On the Husky, the earlier type gearbox is still employed. It has ratios of 1 : 1, 1·516 : 1, 2·330 : 1 and 3·644 : 1, with a reverse of 4·610 : 1 and synchromesh engagement of second, third and fourth gears. A rear axle ratio of 4·55 : 1 is used, while the Lockheed brakes, with 8 in diameter drums, are identical with those of the Minx Special Saloon. The Husky has a wheelbase of 7 ft 2 in, and the front and rear track are respectively 4 ft 1 in and 4 ft 0½ in. Its overall length, width and height are 12 ft 5½ in, 5 ft 0½ in, and 5 ft 1 in, and the ground clearance is 6½ in. The kerb weight is 2,080 lb, appreciably less than that of the Minx.

Rover 80 and 100

FOR 1960, the Rover range, excluding the unchanged 3-litre car, has been reduced from four to two. In place of the previous 60, 75, 90 and 105 models, there are the 80, with a 4-cylinder engine of 2,286 cm³ swept volume, and the 100, which has a 6-cylinder unit with a swept volume of 2,625 cm³. In their general layout and appearance, these new cars are almost identical with their predecessors, with one important exception. This is the adoption of disc brakes for the front wheels; the brakes are of the type introduced as a mid-season modification on the 3-litre model. The 80 and 100 are fitted, as standard, with overdrive, which was previously an optional extra, and neither is available with automatic transmission. In the refinement of their road behaviour and the luxury of their interior, both cars maintain the traditional high standard.

The engine of the Rover 100 is virtually a short-stroke version of the 3-litre unit. It has the same bore, of 77·8 mm, but the stroke has been reduced from 105 mm to 92·08 mm, giving a stroke : bore ratio of 1·18 : 1. Otherwise, the lower half of the engine is unaltered, and its features include a seven-bearing crankshaft, with copper-lead lined bearings, and a chain-driven camshaft mounted relatively high on the side of the cylinder block. The cam followers are of the bell-crank type, and carry floating rollers that bear on the face of the cams.

An unusual arrangement of overhead inlet and inclined side exhaust valves is employed. The inlet valves are actuated by push rods and rockers, while the cam followers operate the exhaust valves directly. Light alloy is employed for the cylinder head, and the head-to-block joint face is

Although the Rover 80 and 100 models are externally the same as earlier cars, each has a new engine, and both are fitted with disc brakes on the front wheels. Overdrive is standard equipment

inclined at 22 deg to the horizontal. The compact combustion chambers are formed by part-spherical pockets at the upper left-hand side of the cylinder bores, bounded by the flat face of the cylinder head and by the pent-roof piston crowns.

On the right of the head is the integral, water-heated inlet manifold, which feeds separate ports; the six-branch exhaust manifold is on the left side. A difference from the 2,995 cm^3 engine is the use of an S.U. $1\frac{1}{4}$ in bore HD6 carburettor in place of a 2 in unit. Also, the air filter and its trunking are altered, to suit the different installation. Otherwise the upper half of the engine, too, is the same as that of the 3-litre car; the reduction of the compression ratio, from 8·75:1 to 7·8:1, appears to result merely from the shortened stroke, since the clearance volumes of the two units are identical.

The maximum gross power output of the new 2·6-litre engine is 104 b.h.p. at 4,750 r.p.m., and the maximum b.m.e.p. and torque are respectively 130 lb/in^2 and 138 lb-ft, at 1,500 r.p.m. For the 3-litre engine, the corresponding figures are 115 b.h.p. at 4,500 r.p.m., 136 lb/in^2 and 165 lb-ft at 1,500 r.p.m. The reduction in compression ratio has inevitably caused a lowering of the b.m.e.p. and of the specific torque output, but these reductions are partially offset by the greater specific power output. This is accounted for by the proportionately larger valves, which have improved the breathing at higher speeds, and so have resulted in an upward extension of the power curve. The actual specific output is 39·7 b.h.p/litre, whereas for the larger engine the equivalent figure is 38·4 b.h.p/litre.

Because of the use of the existing connecting rods with the shorter-stroke crankshaft, the ratio of connecting rod length: stroke is unusually high at 2·24:1. The mean piston speed at maximum b.h.p. is 2,870 ft/min. Expressed in terms of b.h.p/in^2 piston area, the power output is 2·35. In relation to the engine weight and the vehicle kerb weight, the output figures are respectively 0·197 b.h.p/lb and 70·7 b.h.p/ton.

In several important respects, the design of the Rover 80 power unit represents a major departure from the firm's post-war practice in respect of car engines. The most noticeable differences are in the upper half of the engine: there is an orthodox arrangement of overhead valve gear, and the combustion chambers are formed in the cylinder head, which has a horizontal joint face with the block. Previous car engines have had a relatively long stroke but the newcomer has over-square dimensions of 90·47 mm × 88·9 mm, giving a stroke : bore ratio of 0·98 : 1.

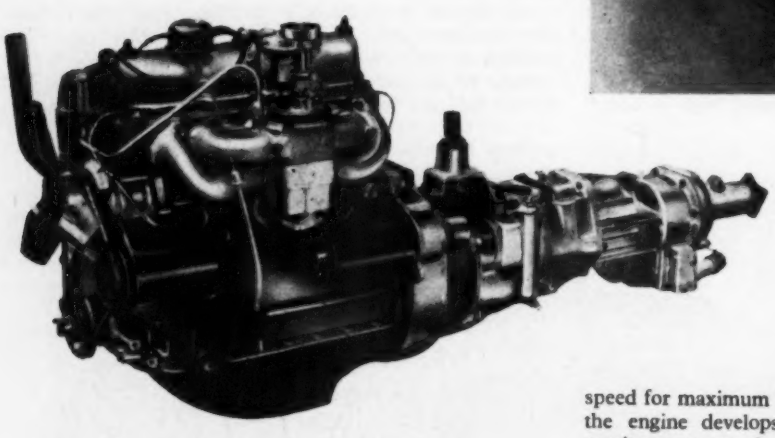
A very robust, counterweighted crankshaft is employed. It has three main bearings, of $2\frac{1}{2}$ in diameter, and the crankpin diameter is $2\frac{1}{4}$ in, so there is considerable overlap on the journals. The main and big-end bearings are lined with copper-lead alloy. As on the 6-cylinder engine, the camshaft is chain-driven and is mounted relatively high in the block, to minimize the length of the push rods. Again, the cam motion is transmitted by rollers, but they are carried in shoe type tappets, reciprocating in slots in steel guides; the lower end of each push rod registers in a socket in the upper face of the shoe.

The cylinder head is an iron casting. It has individual inlet and exhaust ports, on the left side, for all cylinders. The valves are in line and inclined at 10 deg to the vertical; the inlet valves are appreciably larger than the exhaust valves. Cast in the head are combustion chambers of kidney shape. The inclined sparking plugs are disposed on the right, convex sides of the chambers. Each is rather nearer the inlet valve than the exhaust, to provide scouring and cooling by the incoming mixture, and to bring it on to the squish axis. The compression ratio of 7·0:1 is rather low by current standards.

There is a thermostatically controlled hot-spot arrangement between the inlet and exhaust manifolds. A Solex PA 10-5 downdraught carburettor is fitted, and it is fed by an AC mechanical fuel pump. The distributor, driven by a spiral gear on the camshaft, is mounted nearly vertically, mid-way along the block on the right.

For this engine, the ratio of connecting rod length : stroke is 1·97:1, and the mean piston speed at 4,250 r.p.m., the speed for maximum b.h.p., is 2,480 ft/min. At these r.p.m., the engine develops a gross power of 77 b.h.p. The maximum torque is 124 lb-ft at 2,500 r.p.m., equivalent to a b.m.e.p. of 134 lb/in^2. Other indications of the engine's performance are the figures of 33·7 b.h.p/litre, 1·93 b.h.p/in^2 piston area, 0·15 b.h.p/lb engine weight and 52·8 b.h.p/ton vehicle kerb weight.

Both models have an identical layout of the transmission. The Borg and Beck single-dry-plate clutch has a diameter of 9 in and a friction area of 2 × 32·9 in^2. It is coupled to



Left: The Rover 80, 4-cylinder engine. It has over-square cylinder dimensions and a swept volume of 2·3 litres. Above: a disc brake layout identical with that employed on the 3-Litre car has been adopted. These brakes are of Girling manufacture, and the cast iron discs have a diameter of 10·79 in

an orthodox four-speed gearbox, with synchromesh on second, third and fourth gears; the traditional Rover free-wheel is no longer specified. A Laycock-de Normanville overdrive, giving a step-up of 0·778 : 1 and operative on top gear only, is mounted on the rear of the gearbox, which has ratios of 1 : 1, 1·377 : 1, 2·043 : 1 and 3·376 : 1, with 2·968 : 1 reverse. The two-piece open propeller shaft drives a semi-floating rear axle, with a spiral-bevel reduction of 4·3 : 1. In normal top gear, the road speed per 1,000 r.p.m. is 17·8 m.p.h., and in overdrive it is 22·8 m.p.h.

At the rear, the suspension comprises semi-elliptic springs and telescopic dampers. The springs are stiffer than those of the superseded models, and a helper leaf provides a variable rate. At the static load position, the rate is 100 to 110 lb/in, but it becomes 220 to 240 lb/in with the helper leaf in action. The periodicity of the suspension is 88 c/min, and the roll centre height is 9·75 in.

Although the front suspension embodies double transverse links and coil springs, it differs from the majority of such layouts in that the short upper link is of the single, not the wishbone type. The wishbone lower links are unusually long and their pivots are on a bracket attached to the middle of the front cross member of the chassis frame; braking loads are transferred to the frame by obliquely mounted, leading radius arms, and an anti-roll bar is fitted. Seatings for the lower ends of the coil springs are formed in the wishbones, and the upper ends of the springs are housed within the frame side members. Also of telescopic type, the dampers are installed vertically behind the springs. The suspension periodicity is 67 c/min, and the roll centre height is quoted as 2·3 in.

A Burman recirculating-ball type steering box is employed. It has a ratio of 15·6 : 1 in the straight ahead position, increasing to 23·8 : 1 at full lock. The steering linkage is orthodox, comprising a three-piece track rod assembly and a slave lever. For cars equipped with left-hand steering, the positions of the steering box and the slave lever are interchanged. Lock to lock movement of the front wheels requires 3½ turns of the steering wheel, and the turning circle is 37 ft.

The improved Girling braking system is identical with that used on the 3-litre car. It embodies 11 in diameter drums on the rear wheels, with two trailing shoes of 2¼ in width, and discs of 10·8 in diameter at the front. Chromium cast iron is used for the discs, which have a thickness of 0·567 in. The split type calipers of the disc brakes are mounted ahead of the wheel axes; each has two friction pads, having a total swept area of 13·56 in². Vacuum servo assistance is provided, and the reservoir is installed in the valance of the right-hand front wing.

The side members of the chassis frame are of box section. They are bridged by five cross members, all of which, too, are of box section. One is at the extreme front, the second passes beneath the gearbox, and the third is roughly mid-way along the side members: it carries the intermediate bearing of the propeller shaft. The fourth cross member is on the upsweep of the side members, slightly ahead of the rear axle, and the final one braces their rear extremities, beneath the boot.

To minimize the transmission of road-generated noise to the body, rubber mountings are used at its attachment points to the frame. Only one body style is available, a four-door saloon of handsome, conservative lines. A long-standing Rover feature, seldom seen on other cars, is that the doors close on the centre pillar. Thick carpets, leather upholstery and walnut veneer for the facia and door cappings all contribute to the luxury of the interior, and the standard equipment includes a clock, windscreen washer and heater-demister.

The ratio of wheelbase : mean track is 2·14 : 1, and the moderate overhang at front and rear is indicated by the ratio of wheelbase : overall length of 0·623 : 1. For the ratio of mean track : overall width, the figure is 0·793 : 1. Of the kerb weights quoted in the specification data, the distribution between the front and rear wheels is 54·9 : 45·1 for the 80 and 55·8 : 44·2 for the 100, the engine of which is heavier.

Rover SPECIFICATION DATA

Engine—Model 80
Number of cylinders 4
Bore 90·49 mm
Stroke 88·9 mm
Swept volume 2,286 cm³
Maximum b.h.p. (gross) 77 at 4,250 r.p.m.
Maximum b.m.e.p. 134 lb/in²
Maximum torque 124 lb-ft at 2,500 r.p.m.
Compression ratio 7·0 : 1
Crankshaft Three-bearing, forged steel
Valves Overhead, slightly inclined, push rod operated
Combustion chamber type Kidney shape
Carburettor Solex PA 10-5, downdraught
Fuel pump AC mechanica
Dry weight 513 lb, including clutch but without air filter and coil

Engine—Model 100
Number of cylinders 6
Bore 77·3 mm
Stroke 92·075 mm
Swept volume 2,625 cm³
Maximum b.h.p. (gross) 104 at 4,750 r.p.m.
Maximum b.m.e.p. 130 lb/in²
Maximum torque 138 lb-ft at 1,500 r.p.m.
Compression ratio 7·8 : 1
Crankshaft Seven-bearing, forged steel
Valves Overhead inlet, operated by roller type followers, push rods and rockers; inclined side exhaust, operated by roller type followers
Combustion chamber type Compound, formed by part-spherical pocket at side of bore, piston crown and inclined face of cylinder head
Carburettor S.U. HD6 horizontal
Fuel pump S.U. electric
Dry weight 527 lb, including clutch but without air filter and coil

Transmission
Clutch Borg and Beck single dry plate, 9 in diameter, with hydraulic actuation
Gearbox type Four-speed, with synchromesh on second, third and fourth. Laycock-de Normanville overdrive, effective on top gear only
Gear ratios

	Normal	Overdrive
fourth	1 : 1	0·778 : 1
third	1·377 : 1	
second	2·043 : 1	
first	3·376 : 1	
reverse	2·968 : 1	

Propeller shaft Open, two-piece, with needle roller bearing universal joints

Rear axle
Type Semi-floating, with spiral-bevel reduction
Ratio 4·3 : 1

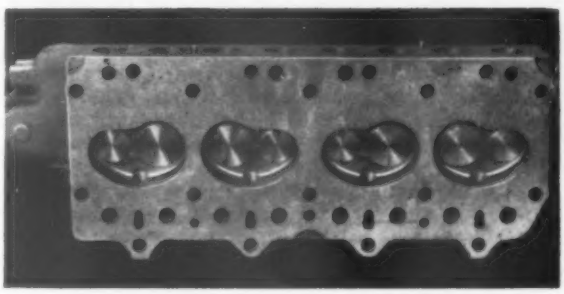
Suspension
Front Double transverse links and coil springs, with telescopic dampers and anti-roll bar
Rear Semi-elliptic springs, with telescopic dampers

Steering
Make and type Burman recirculating-ball
Ratio Straight ahead 15·6 : 1
Full lock 23·8 : 1
Turning circle 37 ft
Turns from lock to lock 3½

Brakes
Front Girling hydraulic, disc type, 10·8 in diameter, with vacuum servo assistance
Rear Girling hydraulic, drum type, 11 in diameter, with vacuum servo assistance and leading and trailing shoes; shoe width, 2¼ in. Mechanical actuation by hand brake
Distribution of braking effort
front 64·3 per cent
rear 35·7 per cent

Wheels and tyres
Wheel type Pressed steel disc, with five-stud attachment
Tyre size 6·00 6·40—15 tubeless
Pressures front 28 lb/in²
rear 24 lb/in²

Dimensions
Wheelbase 9 ft 3 in
Front track 4 ft 4½ in
Rear track 4 ft 3½ in
Ground clearance 7½ in
Overall length 14 ft 10½ in
Overall width 5 ft 5⅝ in
Overall height 5 ft 3⅞ in unladen
Kerb weight, with full fuel tank
80 3,270 lb
100 3,300 lb
Frontal area 20·8 ft²



On the Rover 80 engine, the cylinder head is an iron casting. The combustion chambers are of kidney shape with an inclined roof, and the relatively low compression ratio used is 7·0 : 1

WEATHERPROOFING BODYWORK

Dowty Seloc Washers Seal Against Water or Dust, Protect Paintwork, and Provide Electrical Continuity, While Locking Attachment Bolts or Screws

WITH the greatly increased emphasis on exports since the end of World War II, the problem of efficient car-body sealing has received a good deal of attention. The development of improved materials and designs for the sealing of boot and door openings, in particular, has gone far to meet earlier complaints from users in countries where torrential rain and dusty, unsurfaced roads are normal hazards. Elimination of the main channels of entry for dust and moisture, however, makes it the more important that the many smaller apertures in the body shell, hitherto somewhat neglected, should also receive careful treatment.

That these are more numerous than is generally appreciated is well illustrated by the view of the interior of a Ford Anglia door, Fig. 1. The attachment of door handles, window winders, trim panels and other fittings throughout the bodywork calls for literally hundreds of holes in the steel pressings, each of which should be effectively sealed. In the case of a door, the provision of a winding window and the need for drain holes in the door bottom mean that water and dust cannot be completely excluded from the door shell. Consequently, all the interruptions in the inner skin must be sealed to prevent staining of the trim panel itself and ingress of dust and water to the interior of the vehicle.

Attachment holes for the trim panels and the door handle mountings are easily dealt with. The former are fitted with hollow nylon plugs which are expanded to seal the holes when the studs on the back of the trim panel are inserted. The handle mounting, which pivots, is fitted with a rubber washer. To attach the window-winding gear, to the right of the opening in the centre of the panel, presents a different problem.

This component, shown in larger scale in Fig. 2, is secured by four bolts which are screwed into captive nuts behind the panel. It follows that no sealing washer can be fitted on that side. At the same time, the bolts must be firmly locked against loosening by vibration, which obviates the use of a conventional rubber washer under the bolt heads. A simple, neat and inexpensive solution has been provided by the use of a novel type of washer, known as the "Seloc", which is manufactured by Dowty Seals Ltd., Ashchurch, Gloucestershire. As will be seen from the sectioned examples, Fig. 3, the Seloc consists of a "Shakeproof" washer around which is moulded a casing of synthetic rubber, with a generous overlap of rubber on the toothed periphery. The serrated metal washer is the well-established proprietary article marketed by Barber and Colman Ltd., Marsland Road, Brooklands, Manchester.

The Seloc washer is fitted under the head of the bolt in a similar manner to an ordinary washer. When the bolt is tightened down, as in Fig. 2, the metal teeth cut through the rubber and bite into both the bolt head and the underlying metal panel, providing a vibration-proof lock. At the same time, the compressed rubber firmly seals the gap between the head of the bolt and the panel. Of even greater importance perhaps is the fact that the inevitable "bites" in the paintwork caused by the teeth of the washer are completely sealed from the atmosphere, thus dealing with one of the most important objections to the anti-vibration device where the decorative and protective properties of a paint finish are of importance.

This last point is of particular interest to the motor industry. It offers a complete answer to the problem of securely mounting all types of internal and external fittings and accessories. Among the most common sources of rust are the points where lamps, number plates, windscreen washers, badges, and similar items are bolted to the bodywork. Whether Shakeproof washers are used or not, moisture eventually finds its way under the damaged paintwork round the bolt-holes, and the rust, once started, spreads rapidly. Where the mounting is completely sealed with a

Fig. 1. Interior of a Ford Anglia door. The many holes for the mounting of trim panel, door handles and window winder call for careful treatment to prevent the ingress of dust and moisture to the car

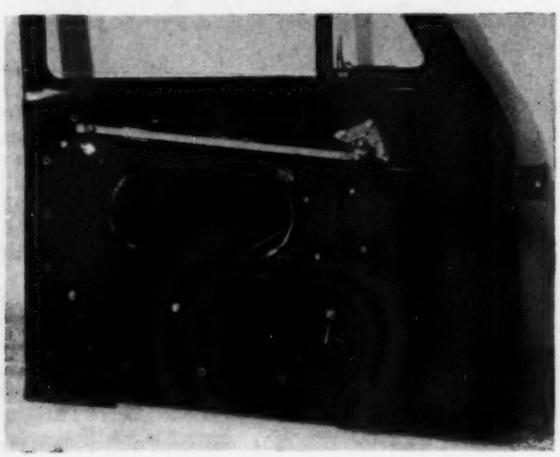
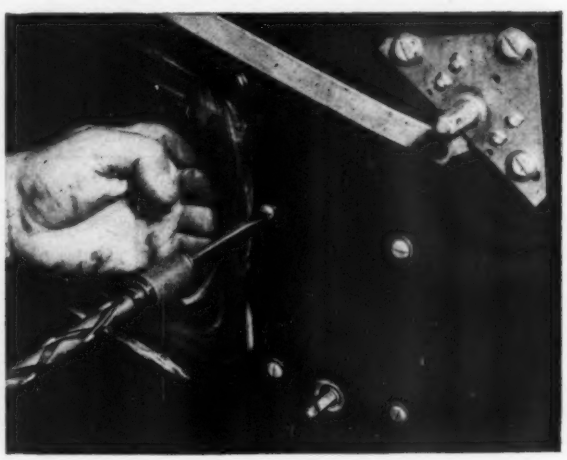


Fig. 2. The window-winding mechanism is attached by four bolts screwed into captive nuts behind the panel. The clearance is sealed and the bolts are locked by Dowty Seloc washers under the bolt heads



Seloc washer this cannot occur, and a permanently secure and rattle-free mounting is obtained at a very low cost.

A seemingly paradoxical feature is of some interest. What looks rather like an insulating rubber washer actually provides excellent electrical conductivity. Where chassis-return wiring systems are in use, or where equipment is to be earthed, the intimate contact between the teeth of the serrated washer and both bolt head and base ensure good electrical continuity. The current-carrying capacity is considerable and, since corrosion is prevented, it does not deteriorate, as is so often the case with conventional methods.

The standard rubber used in these units is impervious to all mineral-base oils, and to alkalis, and they can be used, for example, to provide petrol-proof and fume-proof mountings to petrol tanks. Selocs are approved by the Ministry of Supply, D.E.R.D.(A). Standard types range from 6 B.A. to 1 in diameter bolt size, and all these washers are available with either internal or external teeth.

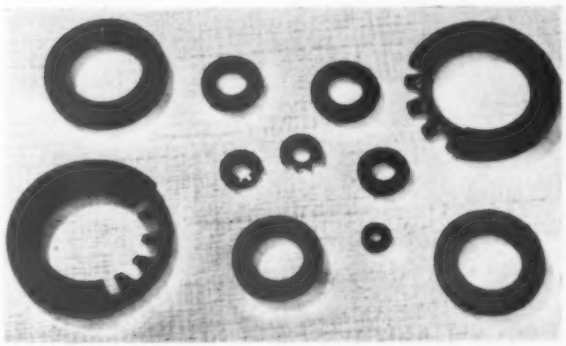


Fig. 3. Typical Seloc washers. The rubber encasing the Shakeproof washer seals the hole and prevents corrosion of the panel where paintwork is scored by the locking teeth as the bolt is tightened

Gauging and Tooling Balls

ALREADY widely used in the U.S.A. and now available in this country, super-precision, master gauging balls are the latest idea in toolroom gauging equipment. The master balls are offered in sizes from $\frac{3}{64}$ in to 1 in diameter in $\frac{1}{64}$ in increments, and they are ideal for setting comparators and other types of gauging equipment, or for use as standards in metrology and inspection departments. The balls are made from chrome alloy steel and hardened to 63-65 Rockwell C. Tolerances are held to $\pm 0{\cdot}000010$ in on diameter and 0·000010 in on sphericity. The surface-finish is 1 micro-inch. Prices range from 30s. each for $\frac{3}{64}$ in diameter balls to 60s. each for 1 in diameter balls. Master balls may be purchased singly or in sets, which are packed in a neat wooden case and include tweezers for handling. As an example of price, set No. 32, which includes 31 balls ranging from $\frac{1}{64}$ in to 1 in diameter in $\frac{1}{32}$ in increments, costs £59 10s. 0d. including case and tweezers. Special sizes can be supplied to order.

Also available in a range of standard sizes is a series of tooling balls. They are precision spheres with an integral shank of suitable diameter. These balls are made from stainless steel hardened to 55-58 Rockwell C and held to a tolerance of $\pm 0{\cdot}0001$ in on diameter and 0·0002 in concentricity. They are intended for use as references in the manufacture, inspection and set-up of jigs, tools and fixtures, and they represent a considerable improvement on the usual toolmakers' "Buttons." Either single pieces or sets suitably packed in wooden cases are available. Further information on sizes and prices can be obtained from the distributors, Insley Industrial Supply Co. Ltd., whose address is 21/22 Poland Street, London, W.1.

Hellermann Polystrap

A BINDING device, called the Hellermann Polystrap, has recently been introduced. This device is designed primarily for use with cable looms, but can be utilized for any tying-up application. It consists of an eye formed by two interlinked holes, one slightly larger than the other, on the end of a beaded tail, four inches long. To use the device, the tail is threaded through the larger hole until the resulting loop is of the required size or tightness; it is then snapped into the smaller hole and thus locked. When it is necessary to remove the strap, for example, to withdraw or add cables, this simple fitting procedure is reversed. In this way the Polystrap may be used again and again.

The strap is moulded in a tough thermoplastic material, which is light and flexible, yet very strong. This material is inert, will withstand temperatures up to 105 deg C, and has excellent electrical insulation properties. It is made in many colours, but the stock shade is white. The Polystrap is available in multiple quantities of 100, the minimum quantity supplied being 100. Samples and prices are available from the manufacturers, Hellermann Ltd., of Crawley, Sussex. The part number is HDP 11113.

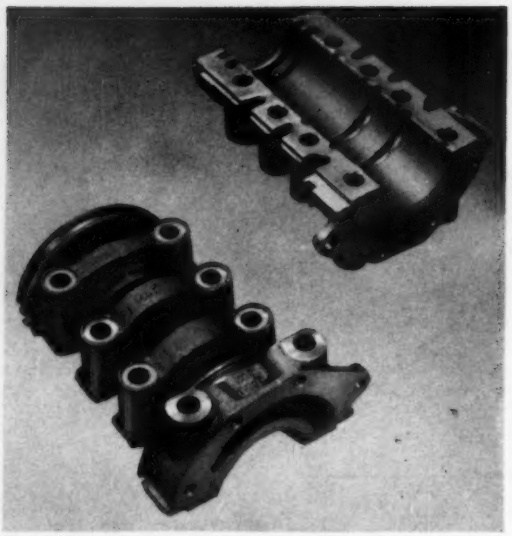
Counting Instruments

A NEW type of multi-circuit batch counter is being marketed by Radiatron, in their IVO range. It has two or four pre-set count positions. Multiple pre-determining counters are, of course, particularly useful where several warning or control signals are required at different stages of a process. The new instruments are available to count shaft rotation, electrical impulses or oscillatory movements. Two or four independent electrical contacts are changed over when the appropriate numbers are reached, and the whole cycle is repeated when a zero set lever is depressed or a detachable key is turned. Electrical reset is also available.

Single pre-determining batch counters are also available and these are made for electrical release, mechanical release, or for giving an audible warning signal only. The single and the double units are available for counting up to 6 digits and the quadruple counters up to 3 digits. In each case the numbers corresponding to the various circuits can be adjusted independently of one another by means of finger-grip wheels.

Another new instrument is a small electrical impulse counter, also an IVO product. This can be built into control panels and is suitable for counting up to 20 impulses per second or even more in special execution.

Its display panel is $2\frac{3}{8}$ in × $1\frac{1}{2}$ in and the counter depth $3\frac{1}{4}$ in. The counter is made for both D.C. and A.C. pulses, the minimum pulse length being 25 milliseconds for D.C. and 33 milliseconds for A.C. A figure of 180 million counts is quoted as the average lifetime of the counter. The instrument is available as a non-reset type or with an instantaneous resetting knob or with a reset key. A unit that will count up to 50 p.p.s, and which has an electrical reset device, is in development. Both counting instruments are available from Radiatron, 7 Sheen Park, Richmond, Surrey, from whom further particulars can be obtained.



MACHINING BEARING CAP CLUSTERS

Features of Caps of European Design Necessitate Departures from Customary American Practice

FOR some considerable period it has been common practice in the U.S.A. to cast a set of crankshaft main bearing caps in a single cluster, to machine the cluster on a transfer line, and finally to gang-saw it into individual components. Usually, the cluster is set up on its side and parallel to the direction of transfer, with either the front or rear end leading. Since European bearing caps, unlike most American designs, require a number of drilling, reaming, tapping, forming and milling operations on the end caps, different methods of handling and machining had to be devised.

The Cross Co. of Detroit, U.S.A., recently designed and built, for a European automobile manufacturer, a thirty-station transfer line for machining clusters of bearing caps at a rated capacity of 436 caps per hour. New techniques, adapted for the machining of the clusters shown at the head of the page, represent a radical departure from conventional practice. They have resulted in simplified machine design, improved operating characteristics and greater reliability in operation. Down time has been considerably reduced. Major changes include machining and transfer with the cluster positioned transversely to the direction of transfer movement; a transfer mechanism of the "lift-and-carry" type; new fixtures and handling devices; an improved method of drilling bolt holes, giving greater accuracy; and a new sawing head and fixture.

Operations that, in the American view, are deemed unusual include the drilling and tapping of a pattern of holes in the No. 1 bearing cap for the mounting of a fan-belt cover; milling the rear face of a lug on the No. 1 cap; and forming felt-seal grooves in both No. 1 and No. 4 caps. On the new Cross "Transfer-matic" line, Fig. 1, these operations are performed by horizontal units, while the drilling

Fig. 1. Cross 30-station "Transfer-matic" line for machining crankshaft bearing cap clusters. Rated output is 109 clusters (436 bearing caps) per hour when working at full efficiency

of bolt holes, and the milling of bearing lock slots, also of the oil-slinger grooves, are effected by vertical units. At the end of the line a special horizontal unit saws the cluster into individual parts.

Sequence of operations

Parts are received on the transfer line with the bearing face and joint face finish broached. The cluster is loaded into the machine with joint face down, No. 1 cap to the left-hand side. At stations 3, 4 and 8, drilling, countersinking and tapping operations on the end of the cluster, No. 1 cap, are completed by horizontal units. Bolt holes are drilled, reamed and spotfaced in stations 3, 4, 7, 8 and 9. Final operations on these holes are performed at station 21, after the part has been turned over.

At station 12, a horizontal single-spindle milling unit, on the left-hand side, face-mills the back side of a lug on the No. 1 bearing cap. Simultaneously, a cross-facing unit generates two felt-seal grooves on the No. 4 bearing cap. A felt-seal groove on the No. 1 bearing cap is machined in two steps by a combination cross-facing unit at stations 15 and 17. Following 180 deg turnover of the cluster from station 19 to station 20, a combination vertical countersinking and gang-milling head countersinks all eight bolt holes on the joint-face side and mills lock slots in all four bearing caps at stations 21 and 22.

Next, two oil-slinger grooves are milled on the No. 4 bearing cap and both sides of the bearing diameter of the No. 2 cap are chamfered at station 25. The cluster is turned over 140 deg from station 28 to station 29, and then lowered, joint face downwards, on to a shuttle-transfer bar mechanism. The cluster is shuttled to station 30, the final station on the line, where it is clamped and then sawed into four separate bearing caps, making up one complete set. As the shuttle-transfer bar carries the cluster into station 30, the bar automatically ejects the preceding cluster into an unloading chute, completing operations on the transfer line.

Bolt holes

In the design of the line, primary attention was given to bolt hole processing, since the size and location of these holes seriously affects the life of the saws employed in cutting the cluster into individual parts at station 30. These saws are the most expensive tools used in machining the bearing caps and, in fact, represent approximately 60 per cent of the total cost of all consumable tooling employed on the machine.

At the sawing station, all bolt holes are located over pins in order to hold the cluster securely during sawing and to prevent movement of individual caps after they have been severed. The slightest movement of the caps results in pinching of the saws as they are withdrawn from the work, and may cause breakage of the carbide tips. For this reason, clearances between the locating pins and bolt holes are critically important. Both bolt hole diameters and centre distances between holes are held within close limits.

In the new machine, this is accomplished by drilling and

Fig. 4. In the final operation, this sawing unit cuts the cluster into individual caps. The cluster is held at 40 deg for maximum chip clearance

Fig. 2. Turnover-transfer mechanism handling the workpiece between stations 19 and 20. A similar device turns the cluster of bearings through 140 deg on the way to the final sawing station

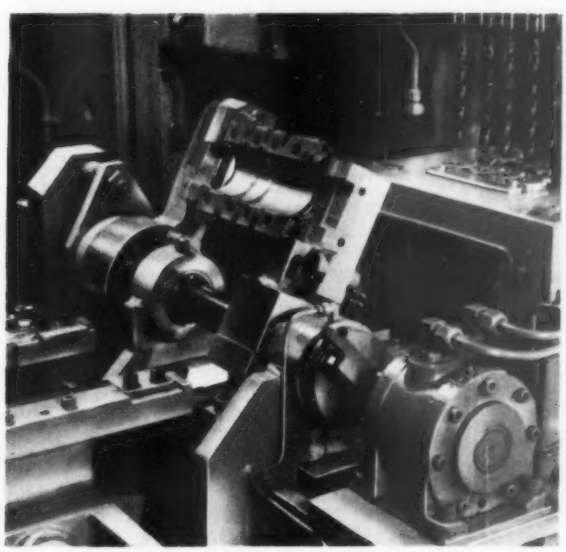
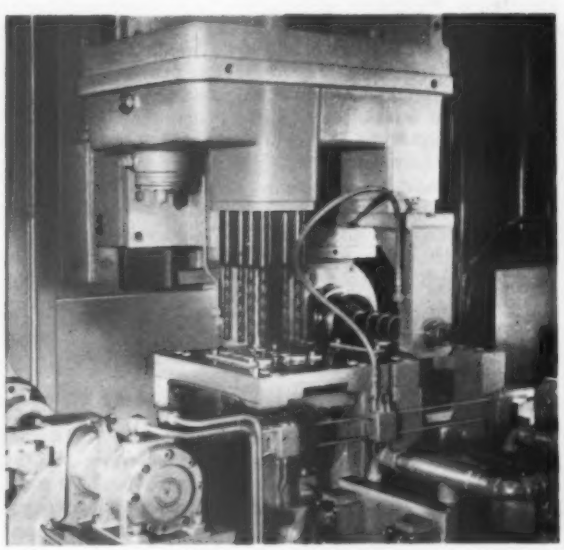


Fig. 3. Combination 8-spindle and gang-milling head for countersinking all bolt holes and milling four bearing lock slots. The head of this interesting machine is powered by a 5 h.p. motor



reaming bolt holes from the side opposite the joint face. With this approach, any drill wander arising from tools that are ground slightly off centre will not affect the centre distance of the holes where the locating pins enter them.

Drilling and milling combination head

An unusual vertical way-type combination head is employed for simultaneous operations at stations 21 and 22. Over station 21, the head carries eight spindles for bolt hole countersinking. Mounted on the head over station 22 is a gang-milling set-up for milling the bearing lock slots.

The multiple-spindle set-up is driven through a gear train in the head, which also powers the drive shaft that rotates the milling cutter arbor. In spite of this unusual combination, the basic head is of the manufacturer's standard design.

Gang sawing unit

Location of this unit represents a change from the position occupied in a conventional lay-out of a bearing-cap line. In this instance it is located off the centre machine base facing the left-hand side of the machine, but with the cutter arbor arranged at an angle of 90 deg to the direction of part flow.

The gang of three saws is powered by an electric motor through a belt-driven gear train. Head, motor and driving mechanism are mounted on the firm's standard No. 128 cross feed unit and a wing base. In operation, the sawing unit cuts the cluster into individual caps as it feeds forward to a positive stop.

Obviously, the location of the sawing unit in this position introduced a novel part-handling problem. This was solved by the development of a new shuttle-type transfer mechanism.

Turnover-transfer mechanism

At the completion of machining operations on the cluster in the last in-line work station in the machine, a combination transfer and turnover device first clamps and then lifts the part out of station 25. It turns the part through 140 deg as it transfers it forward and sets it down on a shuttle-transfer bar at station 29, with the joint surface facing downwards. The shuttle-transfer mechanism consists of a bar located at 90 deg to the direction of part flow in the centre machine base. After the part is placed on the bar, it is clamped and a hydraulic cylinder pushes the bar laterally to station 30, where the cluster is clamped in the sawing fixture.

Sawing fixture

The fixture for holding the cluster during the sawing operation was specially developed for this machine. Mounted on an angular base opposite the sawing head, the hydraulically operated clamping mechanism secures the cluster at an angle of 40 deg to the horizontal. By holding the part up on this angle, the area beneath the clamping mechanism is completely open and maximum clearance is provided for the chips to fall away from the work. The angular mounting arrangement also makes it possible to hold the part relatively close to the sawing head, thereby minimizing overhang of the head during the cycle.

Mechanical Engineering Science

A NEW journal is being published quarterly by the Institution of Mechanical Engineers. It is called the *Journal of Mechanical Engineering Science* and will be available at the following annual subscription rates: to members of the Institution £1 10s. and to non-members £3. In addition, single copies can be obtained at 10s. each by members, and by non-members at £1 each. Normally, there will, of course, be four issues per year, and the December issue will contain the index. In the current year, however, there are only three issues, the subscription rates being proportionately reduced.

The object of the new Journal is to provide an avenue of publication for papers and articles of a fundamental and scientific nature. Its editorial panel has been set up by the Institution. All contributions submitted for publication in the Journal will be examined by the panel, and written contributions on the papers will be considered for publication in subsequent issues. It is intended that the status of the Journal as regards the quality of the contributions published in it shall be equivalent to that of the Institution's proceedings. The panel consider that there is a real need for a publication of this kind which will serve as a focus for first-class scientific papers in the field of mechanical engineering, and they expect that it will quickly establish itself as a recognized forum for such contributions.

Brake Service Guide

A BRAKE service guide has recently been published by Ferodo Ltd. It is a most comprehensive manual describing the operation, identification and maintenance of brakes fitted to all types of British, European, American and Australian motor vehicles. The work comprises 200 pages with 315 line illustrations and graphs, and it has been compiled specifically with the garage proprietor and test fleet maintenance engineer in mind, although it will also be invaluable to instructors on automobile engineering and similar courses.

It describes briefly and simply the main principles governing modern automobile braking systems, and provides sufficient detail to be of everyday assistance to the mechanic. Important sections are devoted to disc brakes and the choice of brake linings in relation to various types of brake drum. Other chapter headings include Relining Shoes, Actuating Systems, Elements of Brake Adjustment, Tracing and Curing Brake Troubles and Brake Testing. This 12 in × 9¼ in book, bound in a flexible hardwearing cover, is obtainable in Great Britain from any Ferodo depot or direct from the head office of Ferodo Ltd., at Chapel-en-le-Frith. Overseas it can be obtained through the Ferodo agent, representative or distributor in the country concerned. The price of the manual is quoted by Ferodo as 40s.

Purolator Filters

A NEW range of pneumatic and hydraulic filters has been introduced by Purolator Products Inc., Rahway, New Jersey, U.S.A. These new filters are known as the P-192 series and are capable of operating at pressures up to 1,000 lb/in² and temperatures up to 275 deg F. They are designed to provide effective filtration for a variety of fluids, including lubricating oils, ethylene glycol, hydraulic fluids and fuels, as well as air and gases. To adapt a filter to handle fluids outside the normal range, it is usually necessary only to change the gasket material. A variety of filter elements can be used, such as paper, wire mesh or the metal-edge type, to deal with particle sizes from 2 to 25 microns. The filter bodies incorporate relief valves, and elements can readily be changed without the necessity of disconnecting the filtration unit from its fluid circuit.

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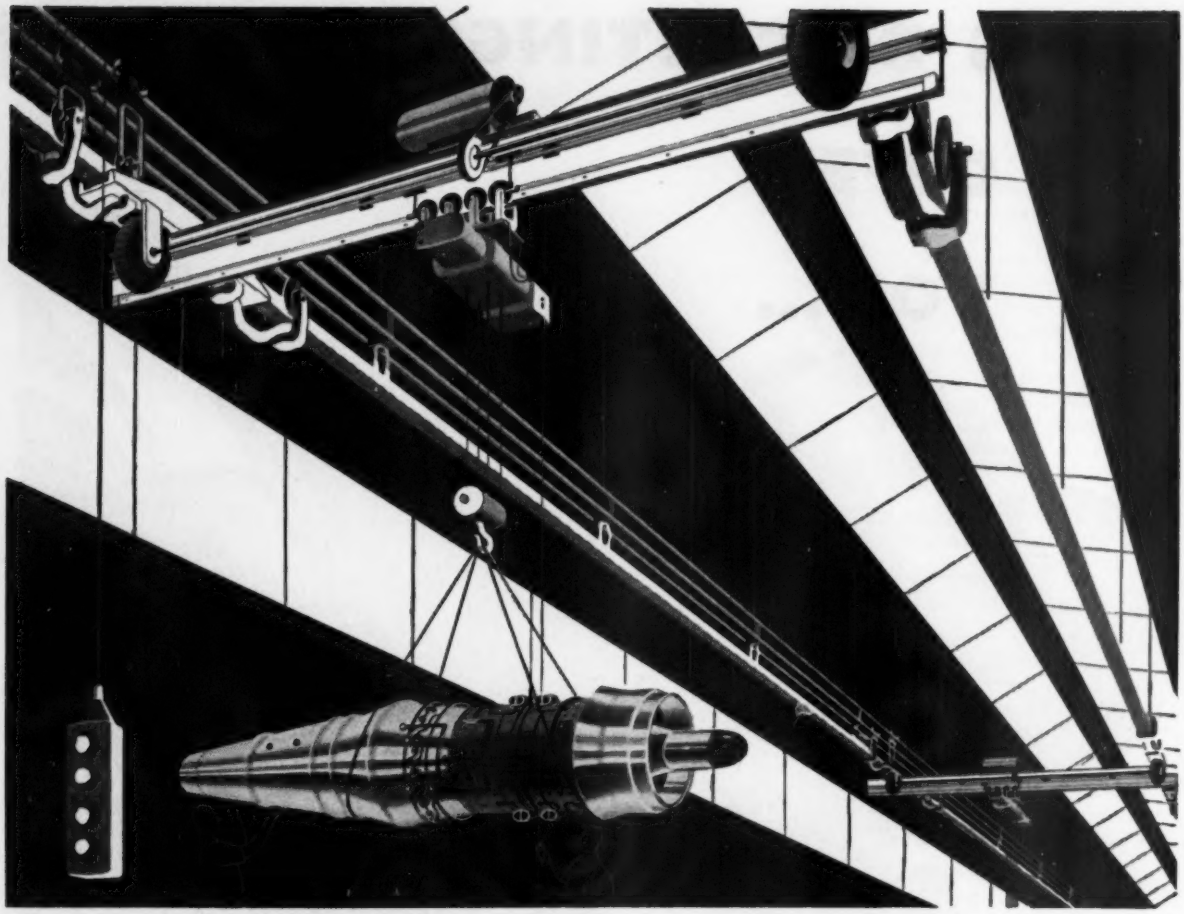

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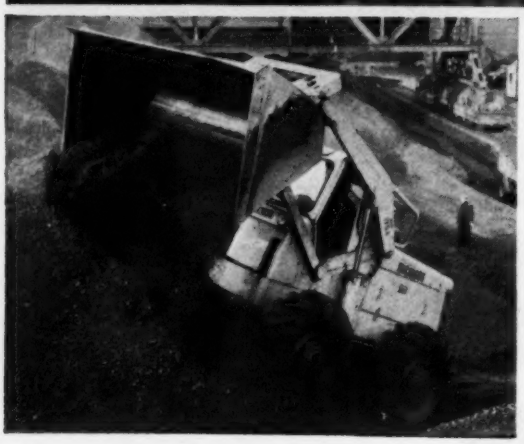
Nylon -



B Nylon S

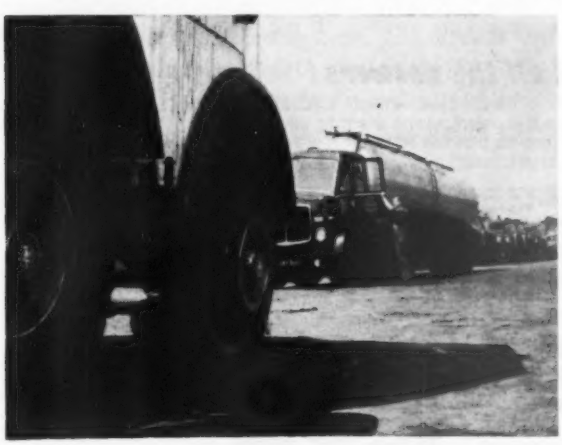
British Nylon Spinners Limited · 68 Knightsbridge · London SW1

takes the load!

Every day, thousands of Londoners pause a moment to watch these huge 'muck-shifters' at work on the South Bank site, clearing the ground for what will, when completed, be London's tallest building. What these spectators do not perhaps know is that they are witnessing yet another triumph for nylon. The huge tyres fitted to these machines, like those on nearly all others of their kind, are nylon-corded. The reason is quite simple —only nylon is strong enough to take the strain.

Nylon is the toughest tyre-cord material in use today. For both the big fleet operator as well as the 'muck-shifter', nylon is setting altogether new standards for reliability because it combines great strength with tremendous powers of energy absorption. Nylon also stands up superlatively well to the bending and twisting to which all tyres are subjected. This flex resistance means that nylon-corded tyres can keep going for tens of thousands of miles and still come back for more.

It's *Nylon* for lasting strength



And here are three more reasons why big transport companies are finding that it pays them well to switch to nylon-corded tyres:—

*** Nylon means more remoulds**

Nylon's outstanding strength, resilience and toughness mean that nylon-corded tyres can be re-moulded time and again—in this way tyre life is multiplied many times.

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Nylon-corded tyres are unaffected by water seepage through casing cuts and give the same high performance, wet or dry.

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Nylon cords generate less heat than any other tyre cord and run cooler on long journeys.

For a modest premium nylon-corded tyres can save you time, money and frustrating delays. For more information about them, please get in touch with your usual supplier. He will be happy to tell you *all* about them.

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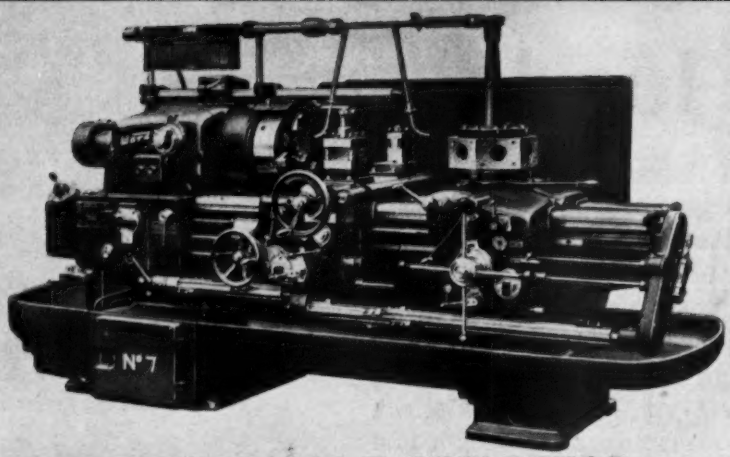

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Spindle: Mounted in ball and roller bearings.

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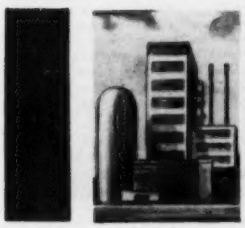

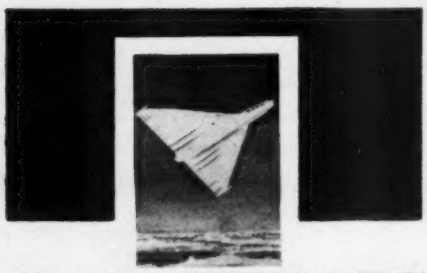

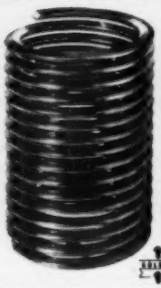

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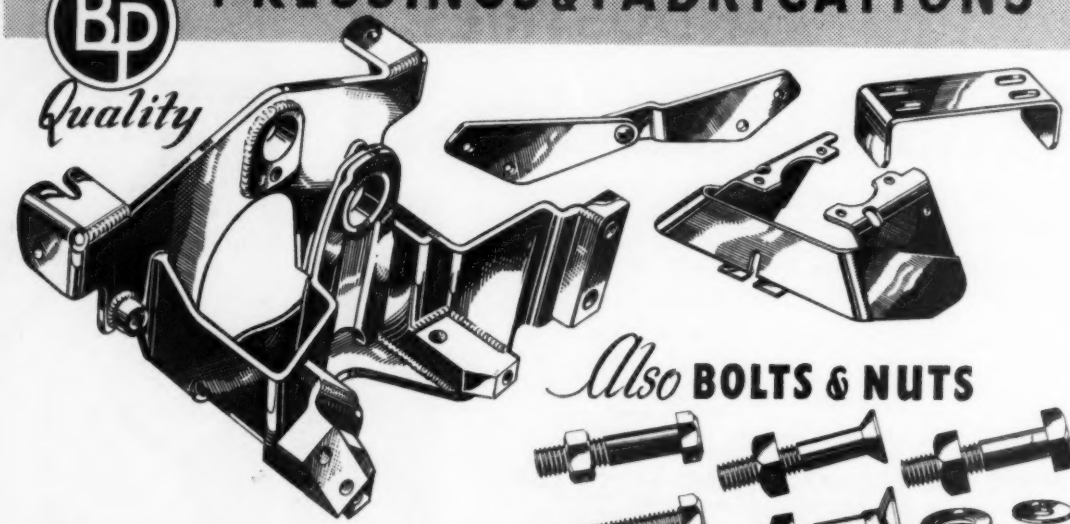

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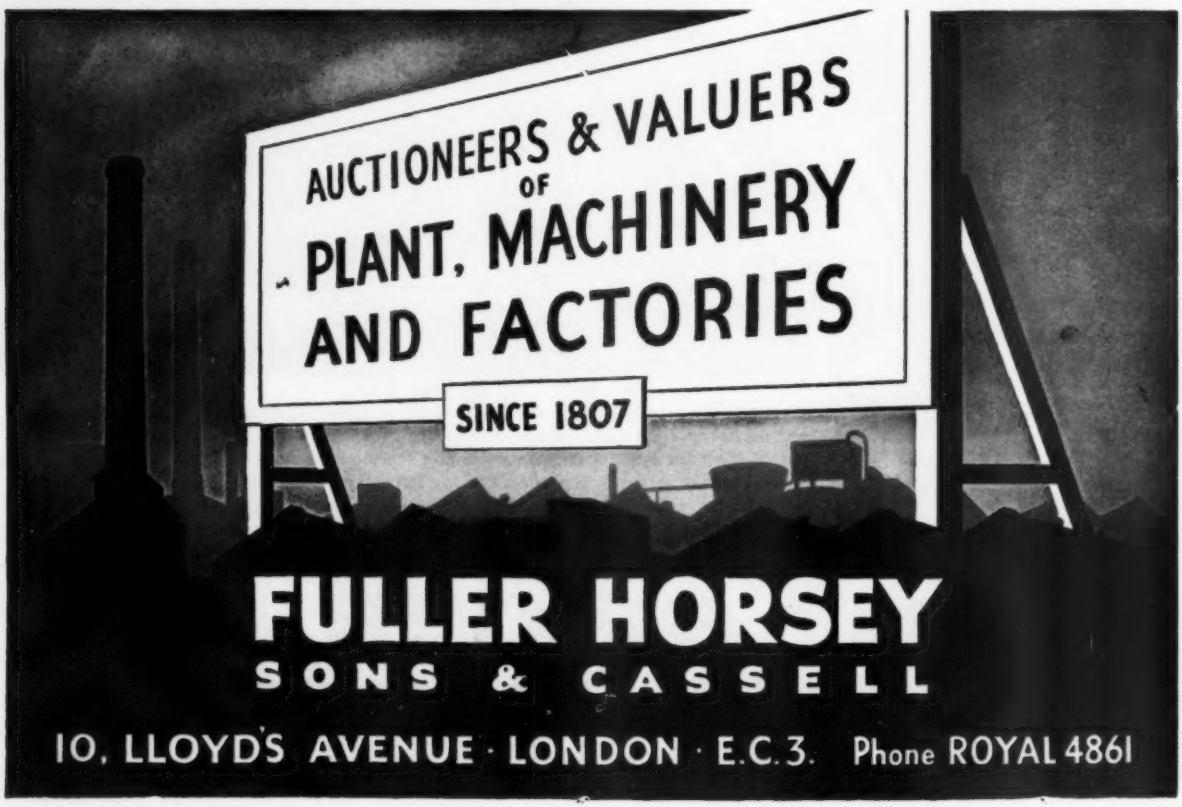
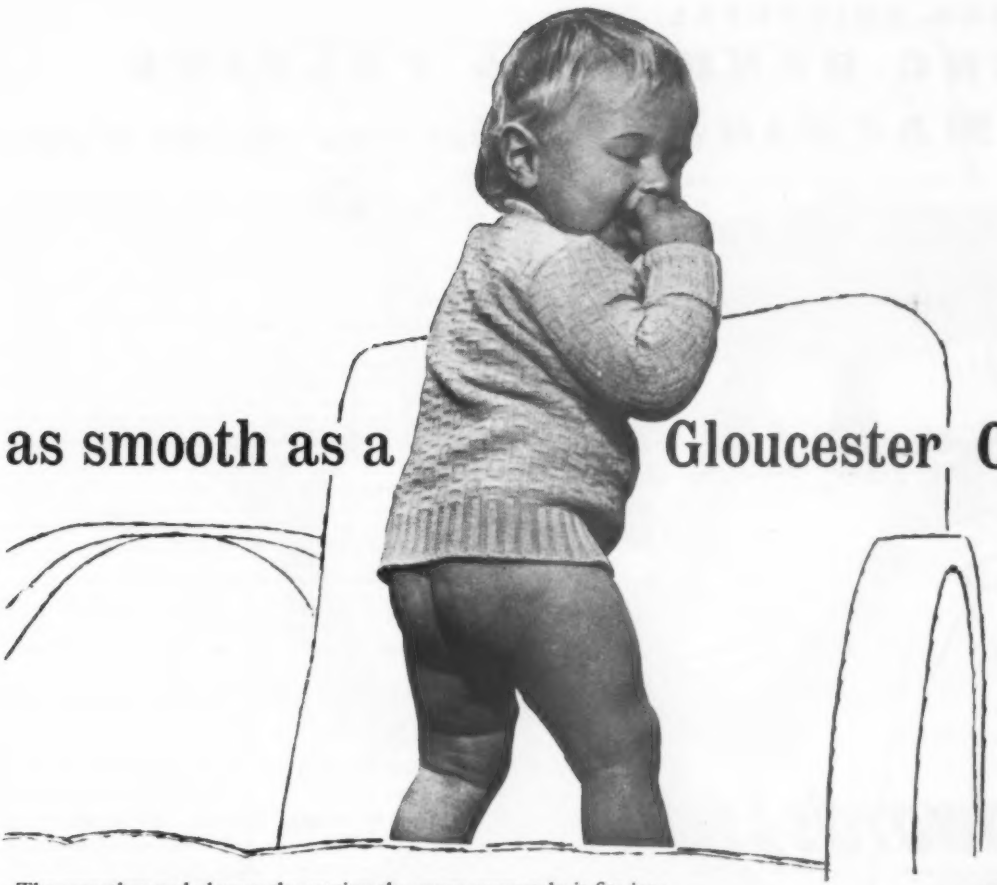


Rathbone

as smooth as a Gloucester Casting



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Vehicle pinion housing in Malleable Iron. Weight 28 lbs.

A typical Gloucester Malleable specification.

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Elongation	18%	Elongation	5%
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At the moment, the 6ft. 2in. machine can be delivered ex stock.

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3

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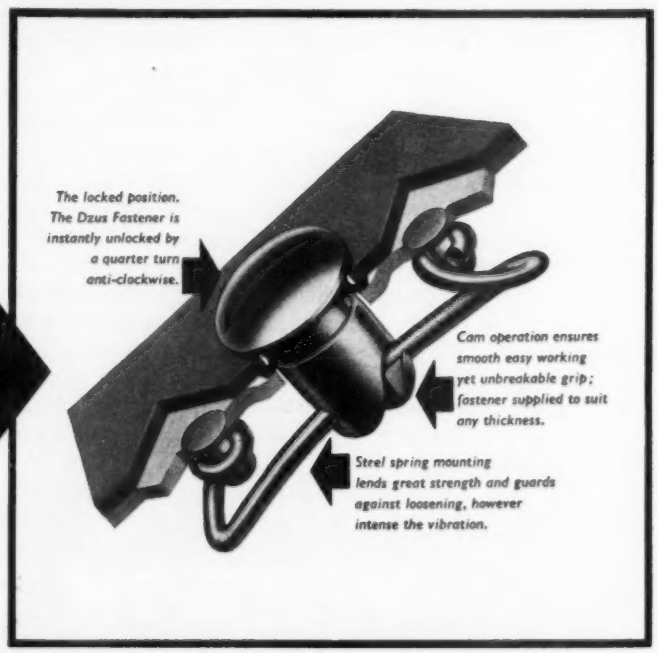
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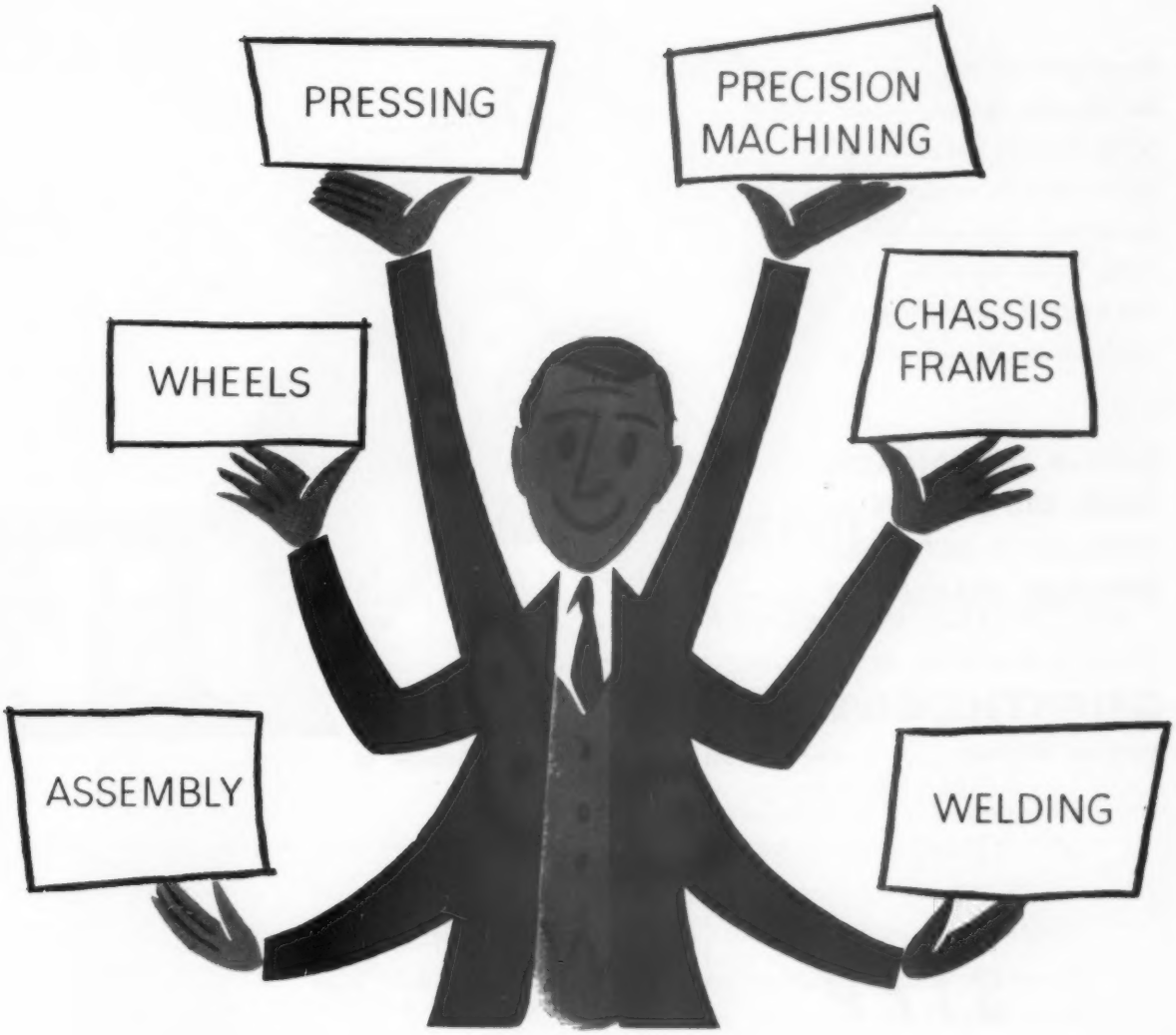

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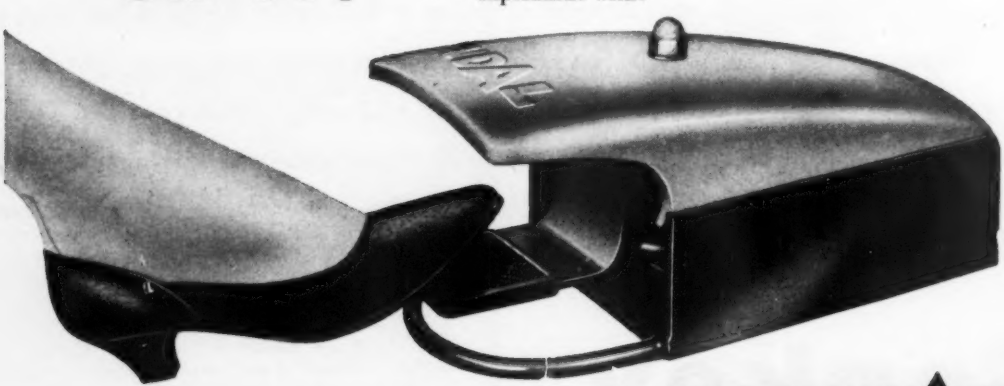

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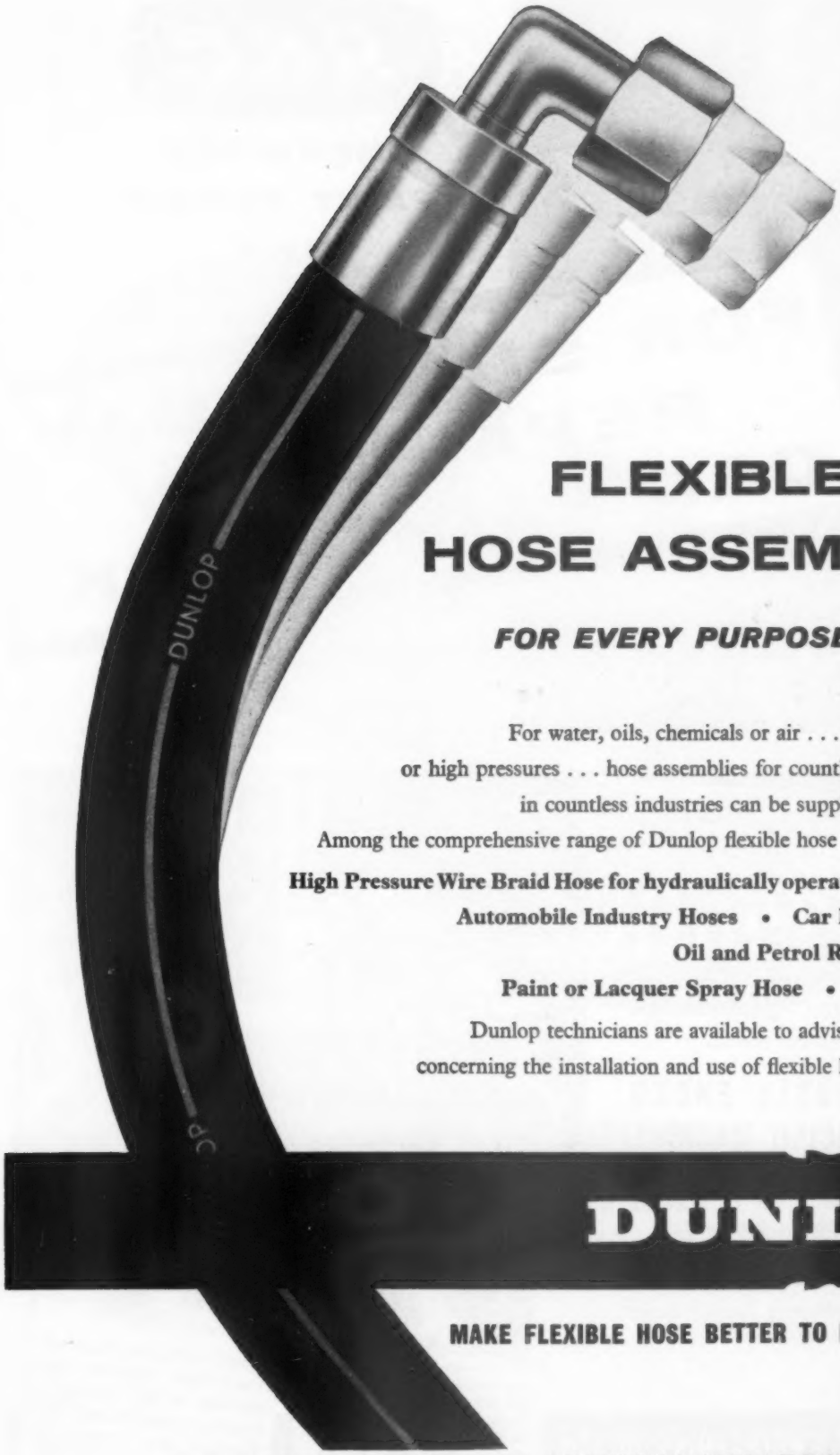

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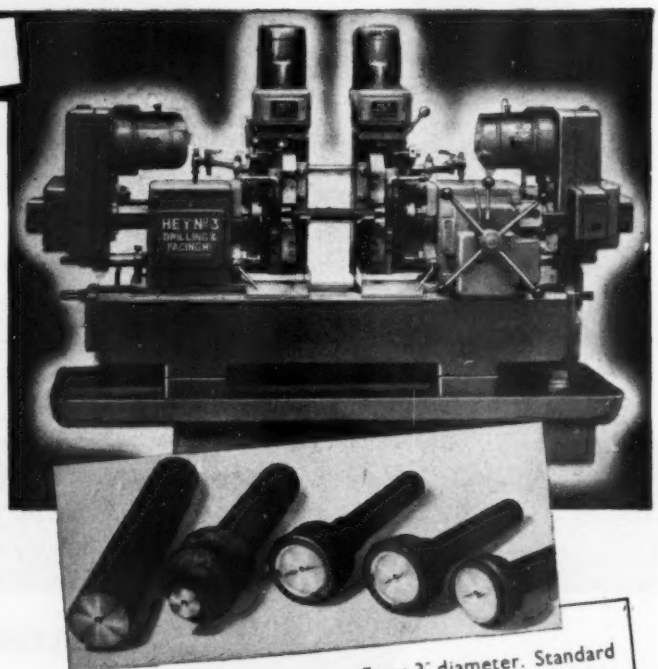

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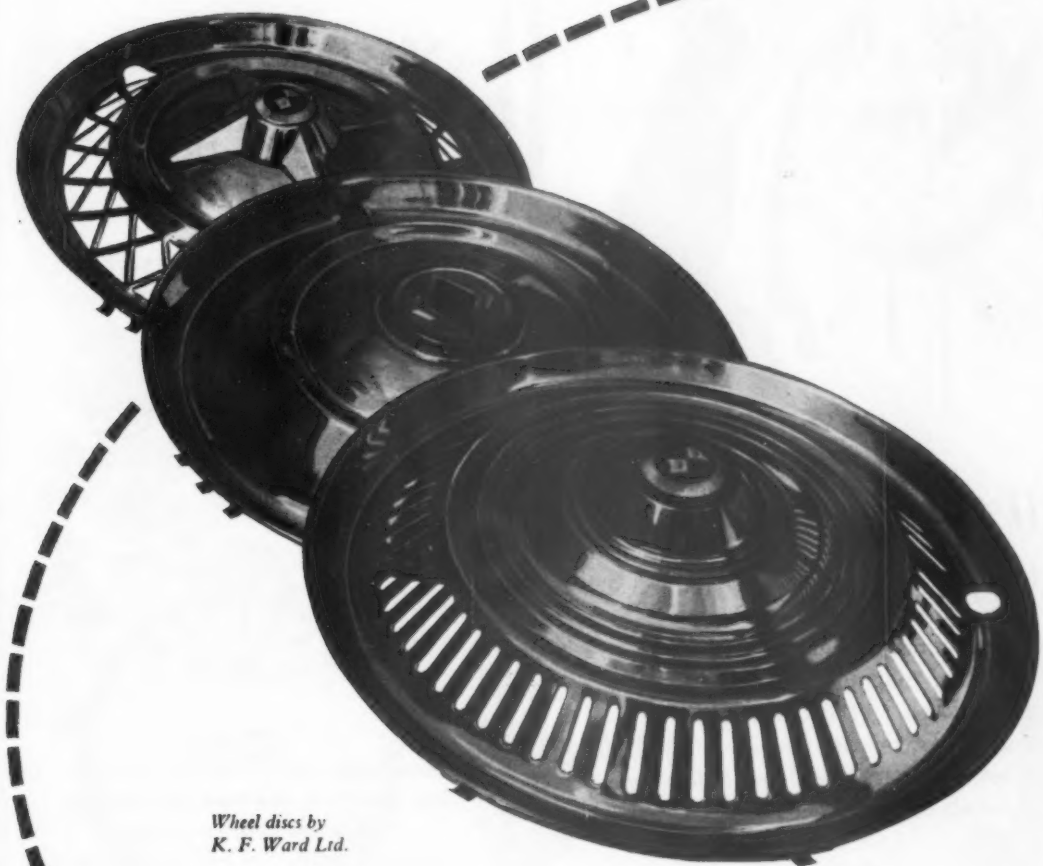

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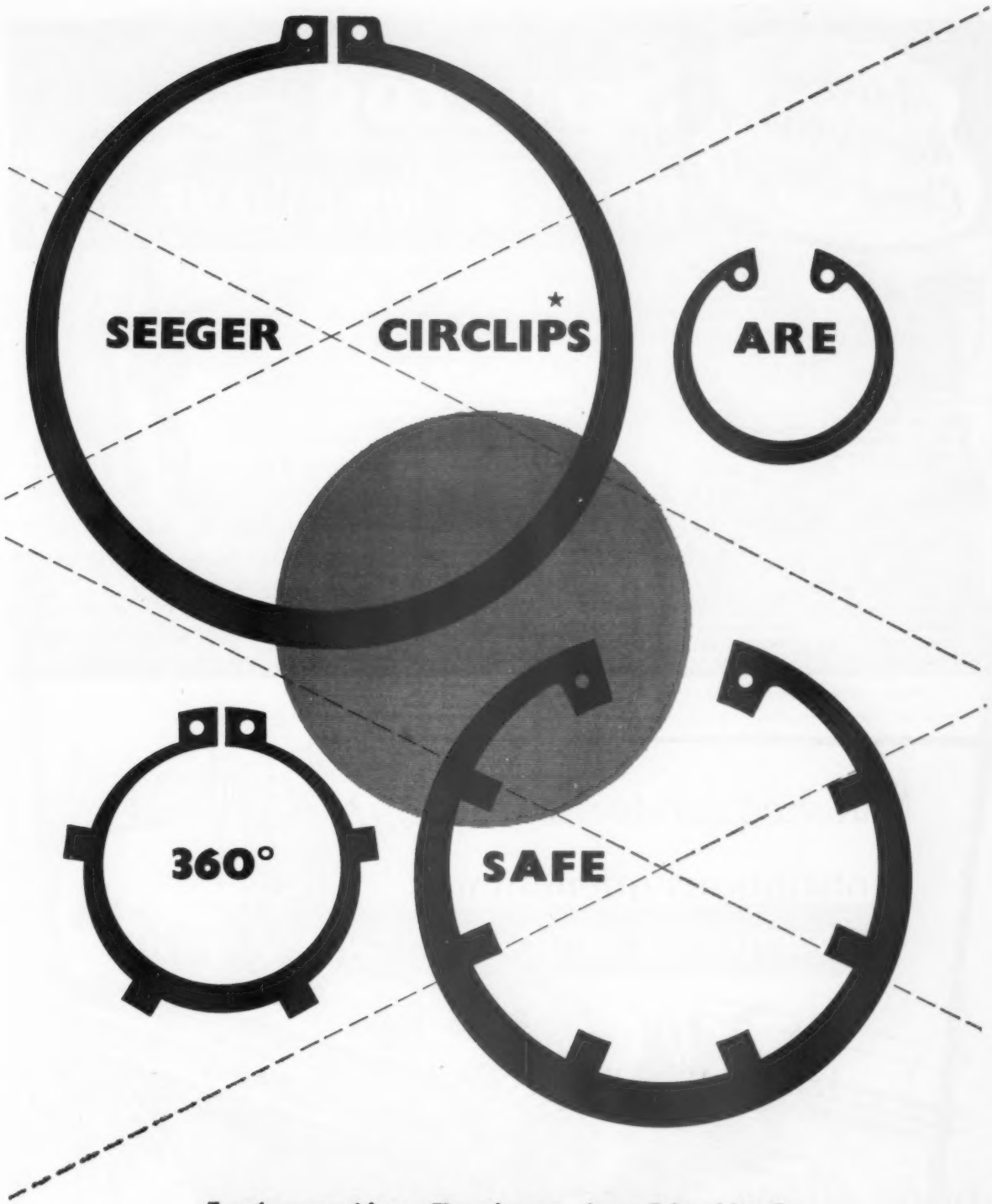

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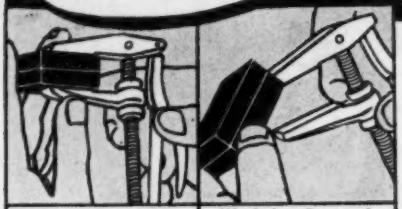
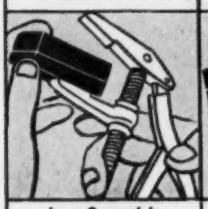


Lower Jaw Adjusted to Correct Distance

Workpiece Removed, Lower Jaw Adjusted for Pressure



Jaws Opened by Trigger Handle to Insert Workpiece

Trigger Handle now Locks Jaws on Workpiece

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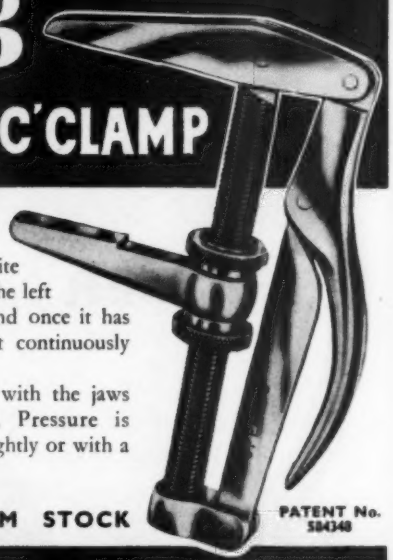

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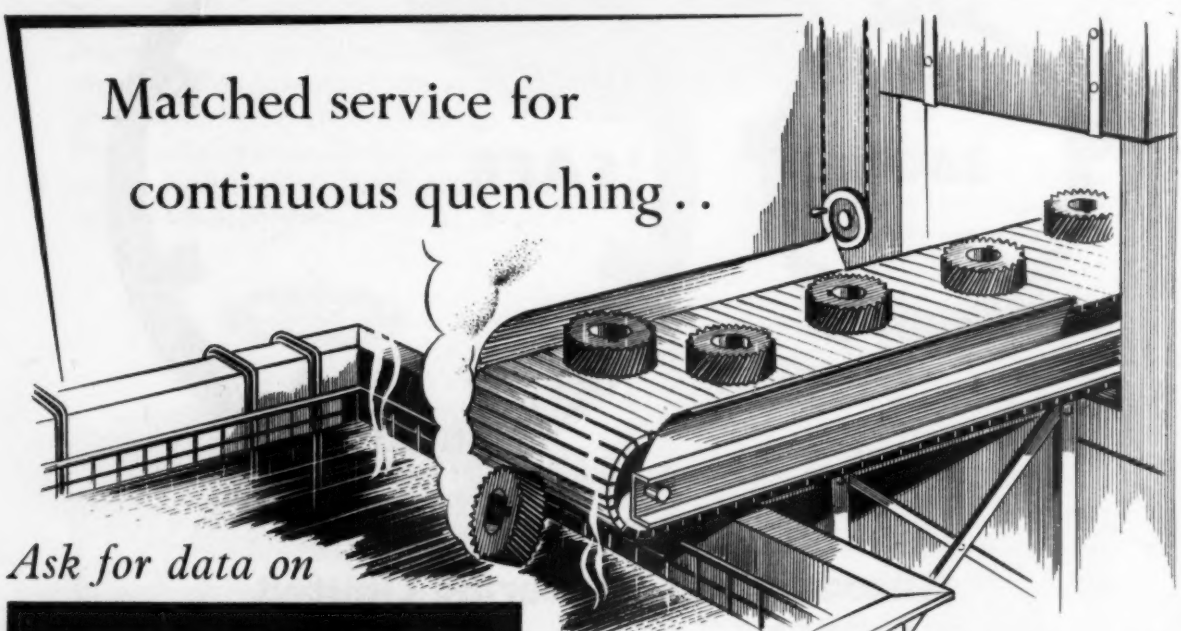

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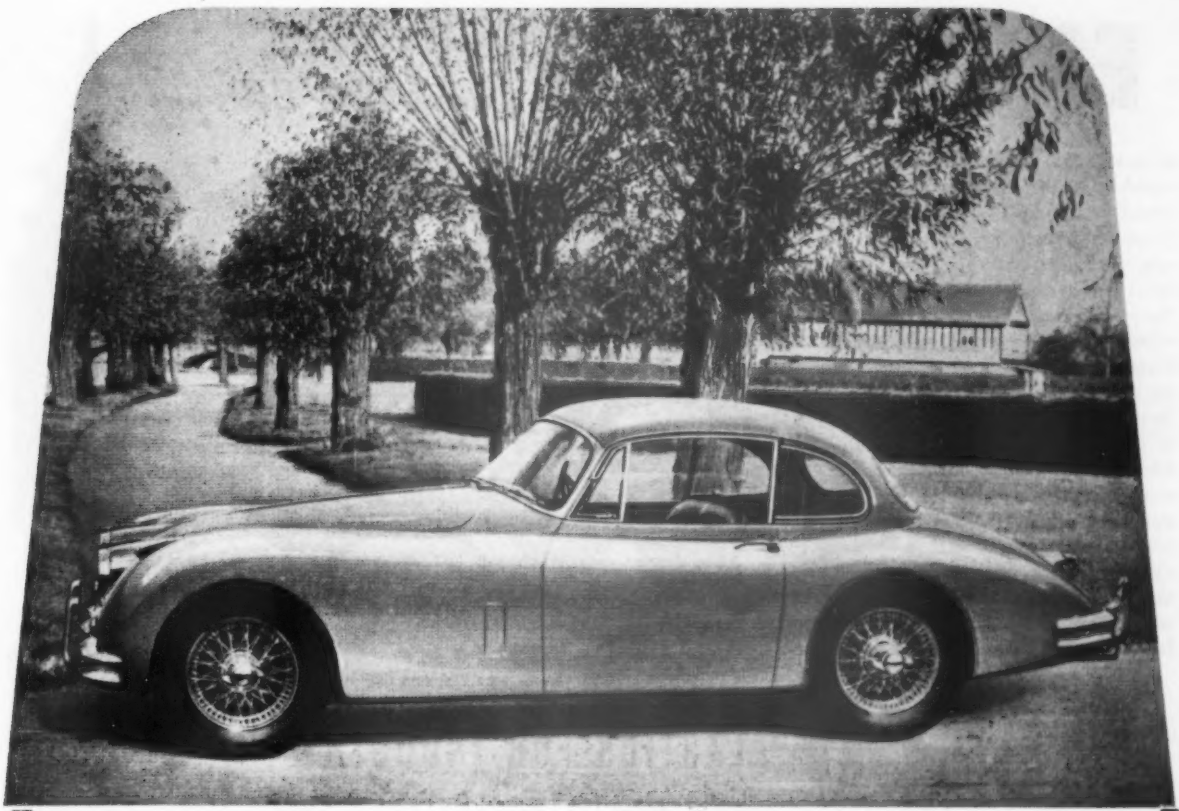

Photograph of the new XK150 by courtesy of Messrs. Jaguar Cars Limited

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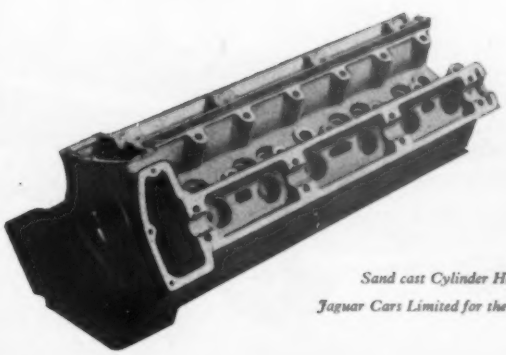

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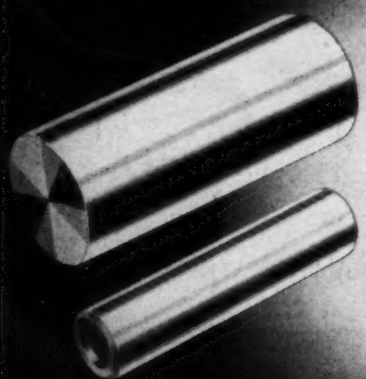

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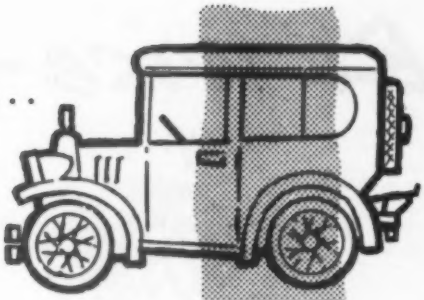

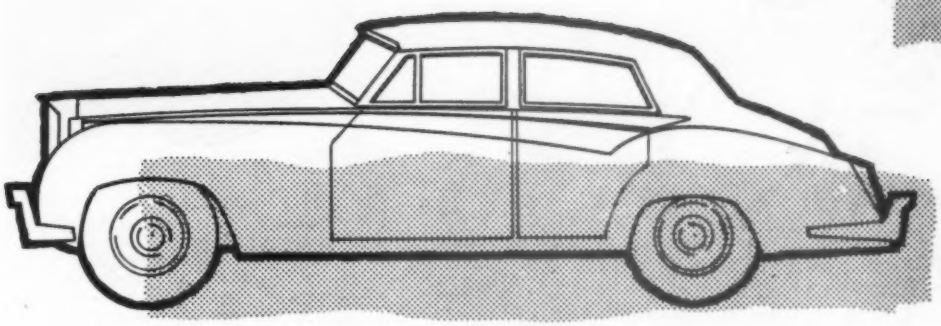

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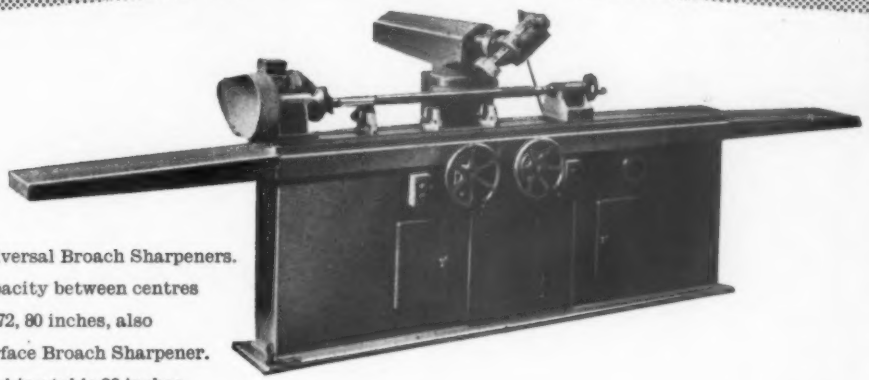

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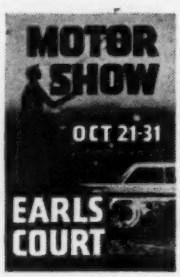

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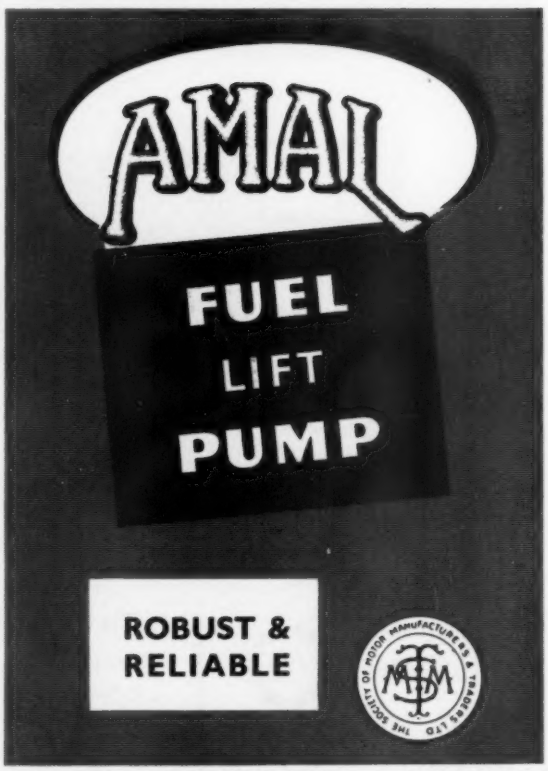

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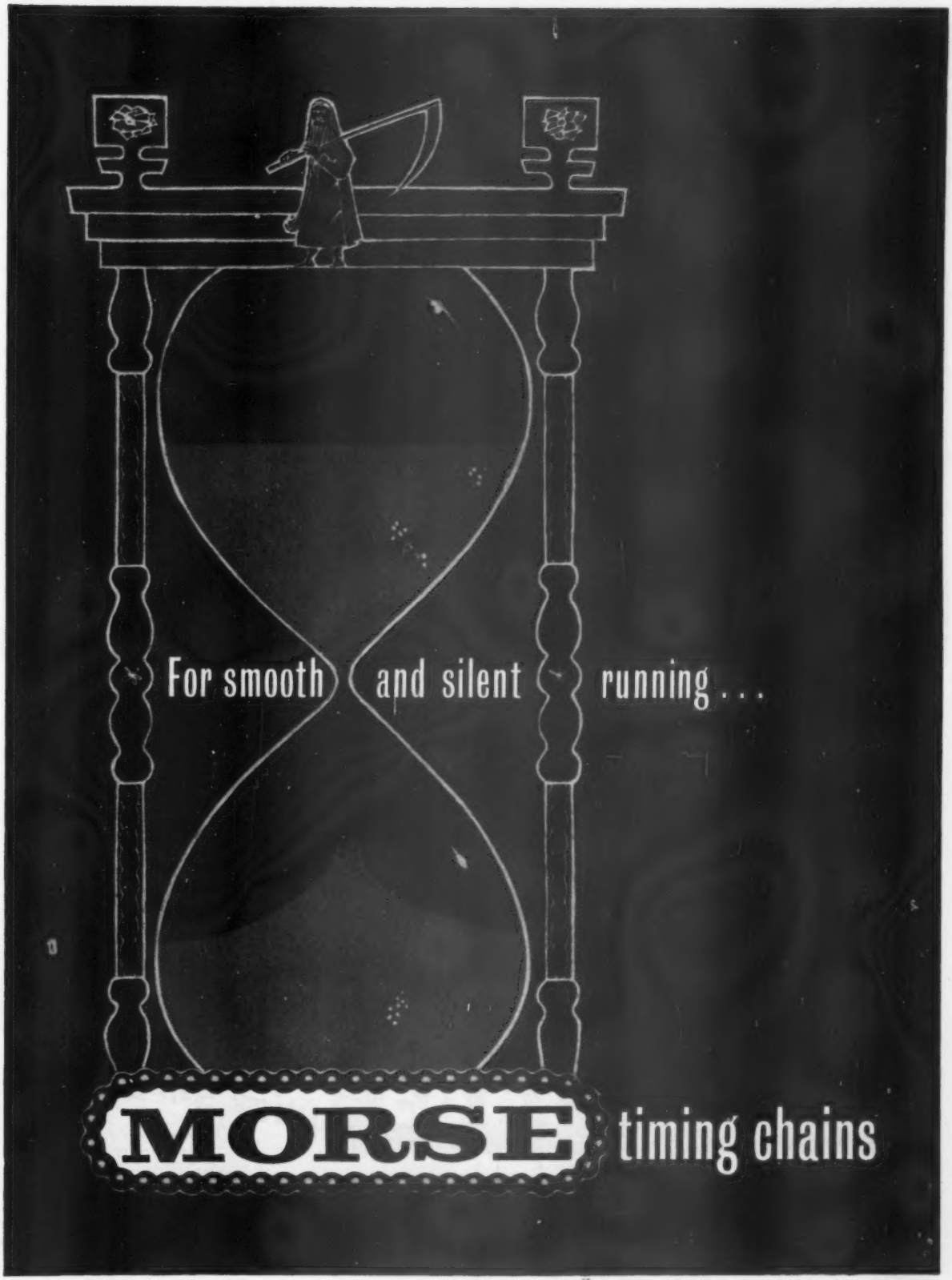

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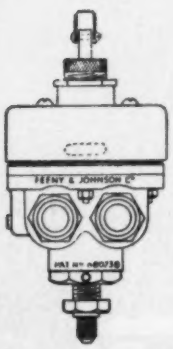

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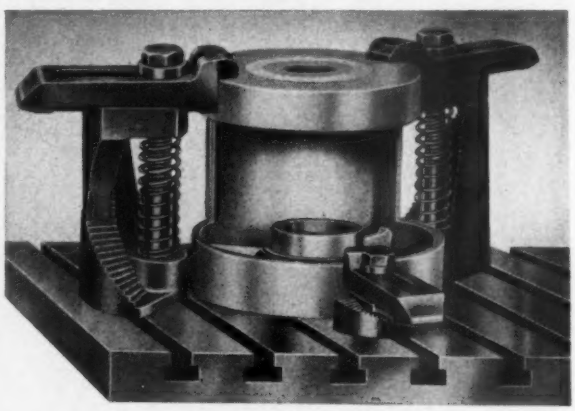

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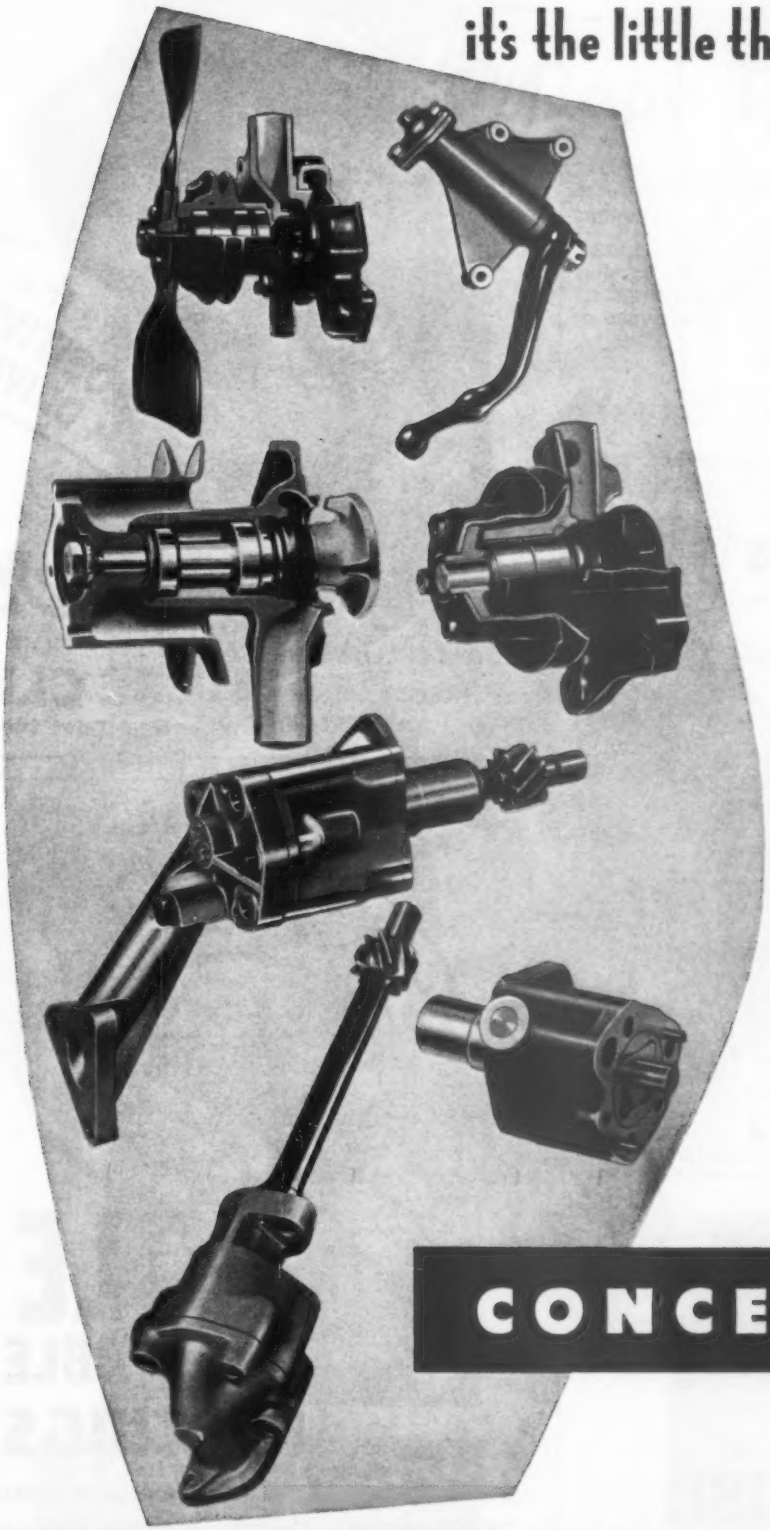

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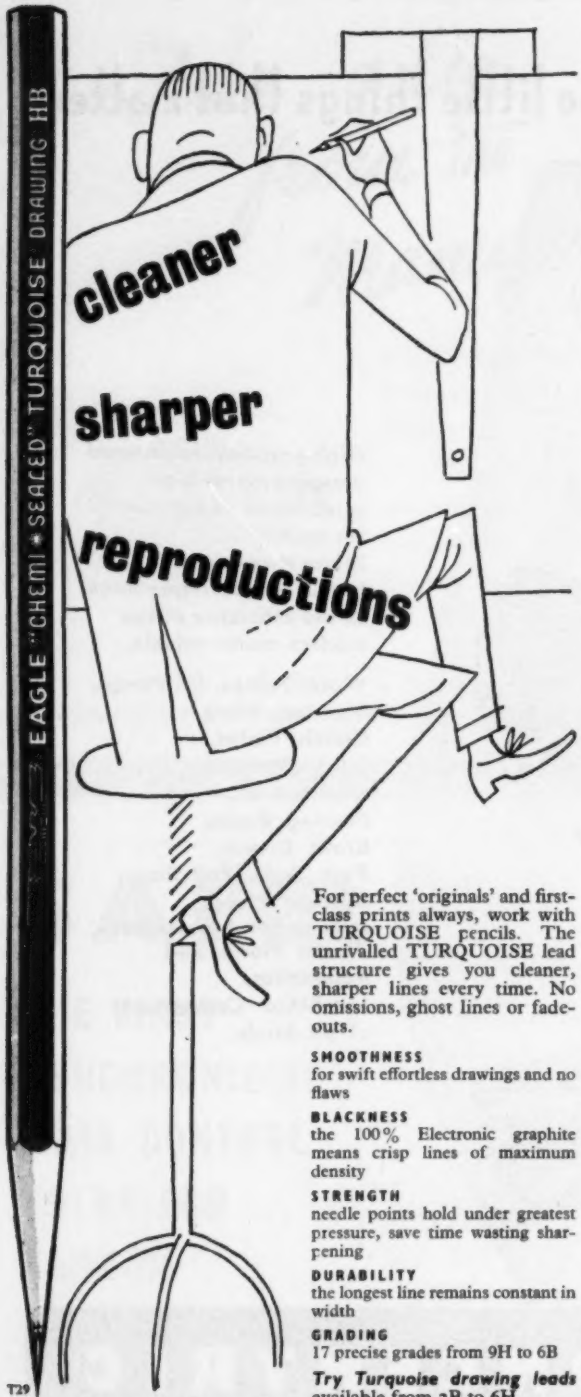

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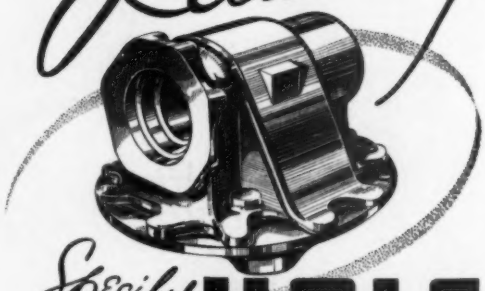

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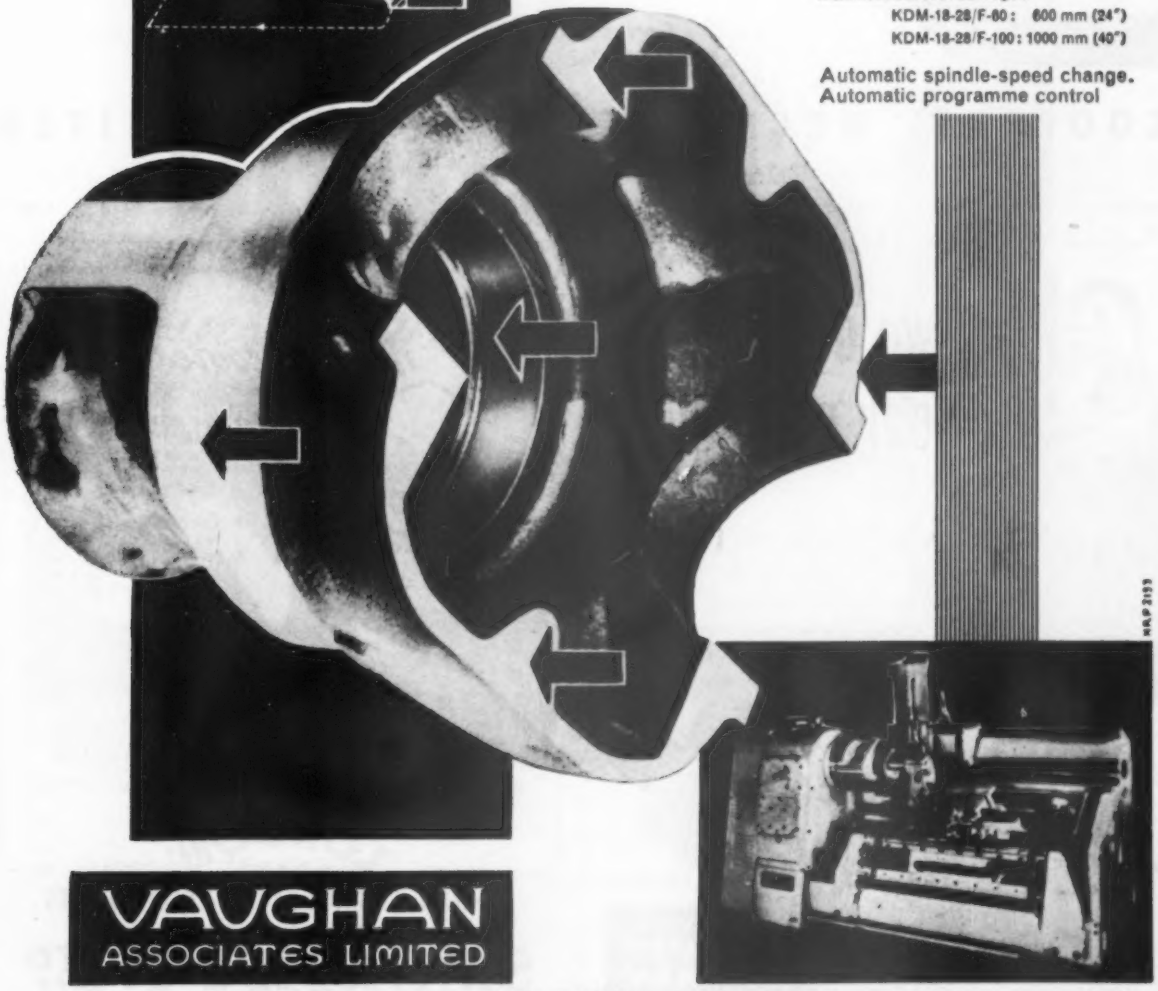

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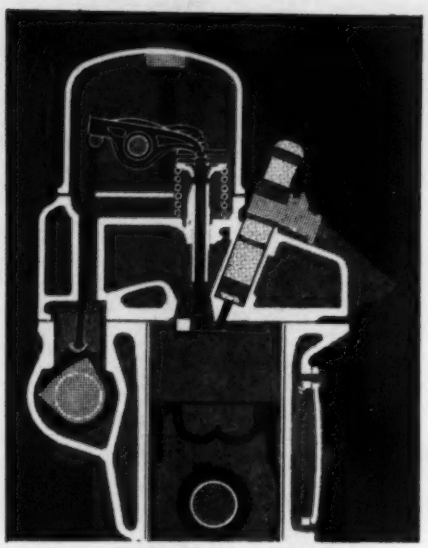

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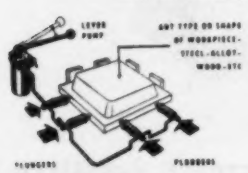

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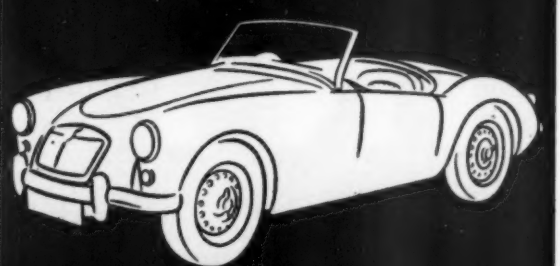

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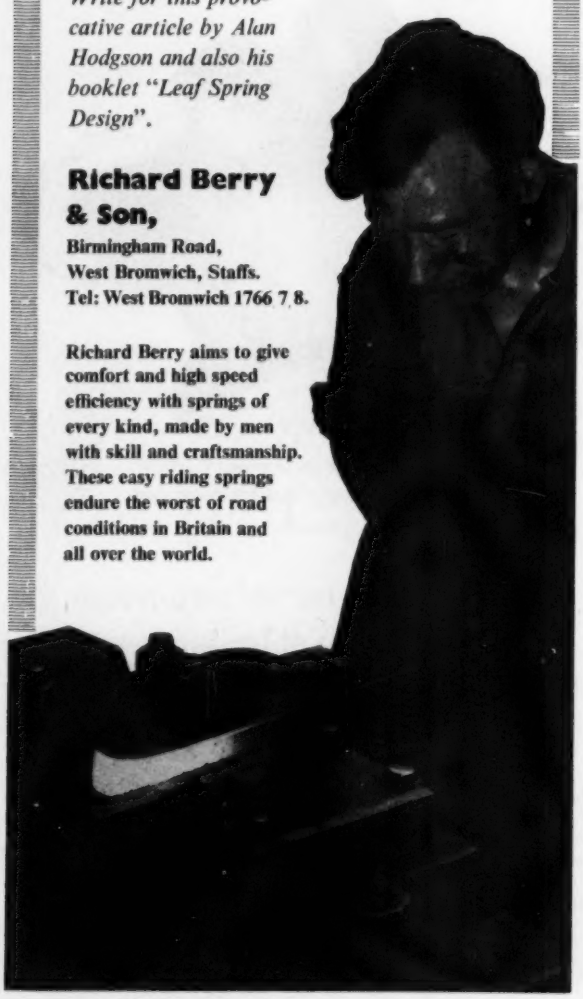

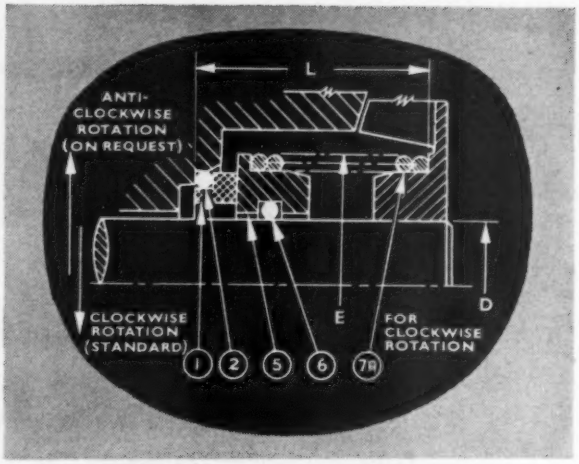

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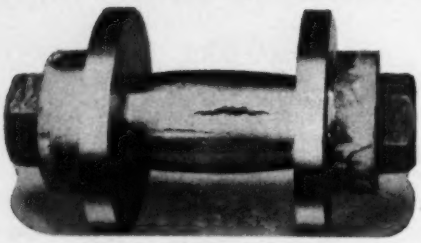

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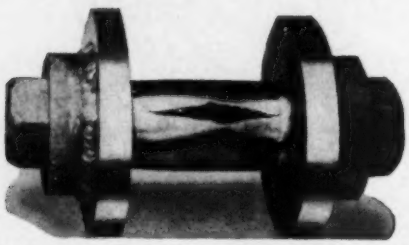

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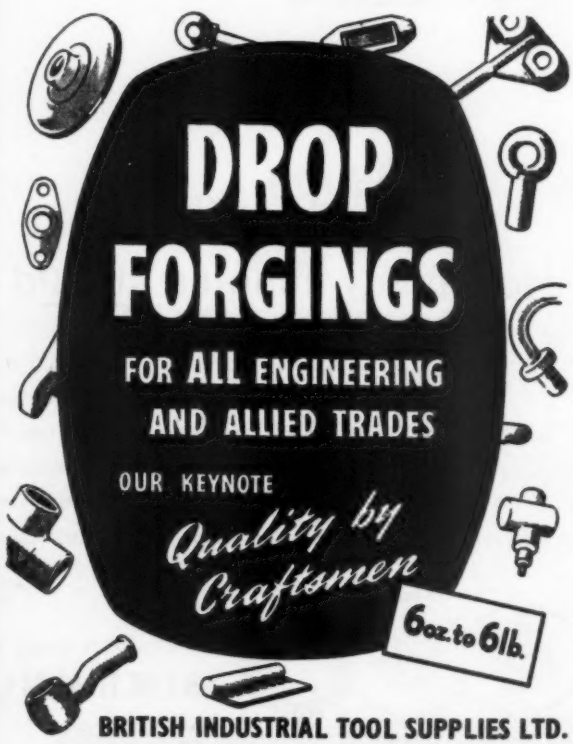

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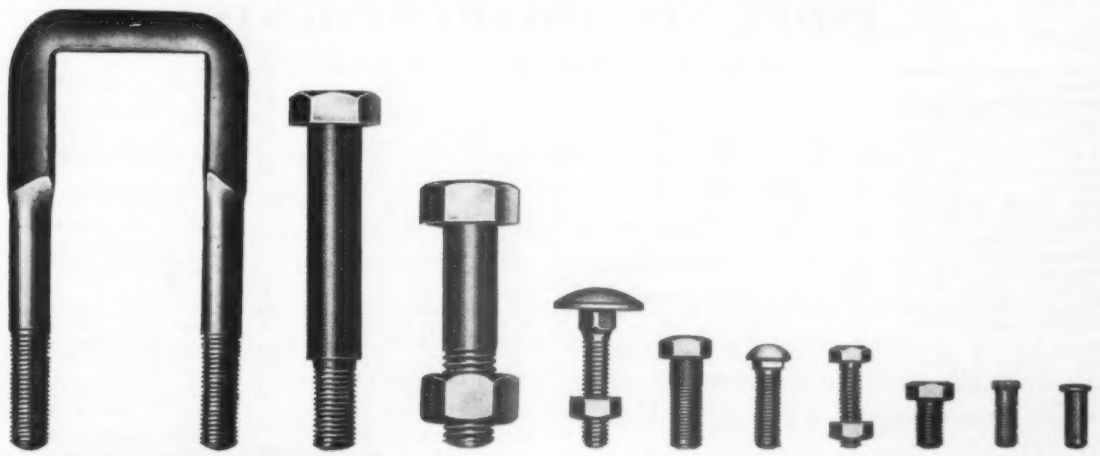

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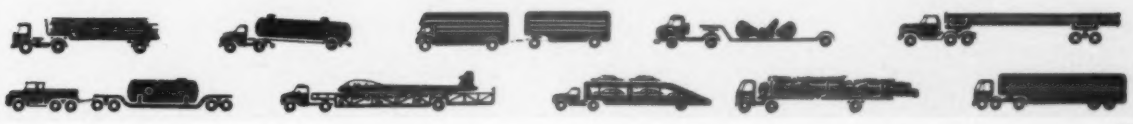

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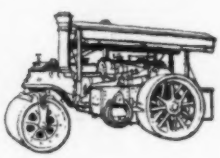

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